NASA CR-121193
APS-5404-R
Volume II

SMALL, HIGH-PRESSURE RATIO COMPRESSOR MECHANICAL ACCEPTANCE TEST

by

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AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA Lewis Research Center
Contract NAS3-14306

Requests for copies of this report should be referred to:

National Aeronautics and Space Administration
Scientific and Technical Information Facility
P. O. Box 33
College Park, Maryland 20740

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA CR-121193		

4. Title and Subtitle
SMALL, HIGH PRESSURE RATIO COMPRESSOR MECHANICAL ACCEPTANCE TEST

5. Report Date
June 1973

6. Performing Organization Code

8. Performing Organization Report No.
APS-5404-R, Vol. II

10. Work Unit No.

11. Contract or Grant No.

13. Type of Report and Period Covered
Contractor Report

14. Sponsoring Agency Code

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AiResearch Manufacturing Company of Arizona
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12. Sponsoring Agency Name and Address
National Aeronautics and Space Administration
Washington, D.C. 20546

15. Supplementary Notes
Program Monitor, Robert Y. Wong, NASA Lewis Research Center, Cleveland, Ohio

16. Abstract

The Small, High-Pressure-Ratio Compressor Program was directed toward the analysis, design, fabrication and mechanical testing of a centrifugal compressor for a pressure ratio of 6:1 and an airflow rate of 2.0 pounds per second.

The final report is in two volumes. The first volume (NASA CR-120941) covers the analysis, selection, and design of the compressor and research package. The second volume (NASA CR-121193) covers fabrication and mechanical testing of the research package.

Mechanical testing was performed to demonstrate overspeed capability, adequate rotor dynamics, electrical isolation of the gas bearing trunnion mounted diffuser and shroud and the effect of operating parameters (speed and pressure ratio) on clearance of the compressor test rig. The speed range covered was 20 to 120 percent of rated speed (80,000 rpm). Following these tests an acceptance test which consisted of a 5 hour run at 80,000 rpm was made with approximately design impeller to shroud clearances.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Compressor/Impeller High Pressure Ratio	Unclassified-unlimited

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	98	$3.00

* For sale by the National Technical Information Service, Springfield, Virginia 22151

FOREWARD

This is Volume II of the final report covering the design, fabrication, and mechanical tests performed under contract NAS3-14306 during the period of August, 1971, through November, 1972.

This contract with AiResearch Manufacturing Company, Phoenix, Arizona, was under the technical direction of Mr. R. Wong, Lewis Research Center, of the National Aeronautics and Space Administration.

Mr. G. R. Metty and Mr. W. I. Shoup conducted the mechanical test program under the direction of Mr. K. W. Benn and Mr. J. T. Irwin at the AiResearch,Phoenix, test laboratory.

The efforts of the following men were greatly appreciated in the conduct of the program:

Mr. R. Jonas - Instrumentation Engineering

Mr. G. L. Reese - Materials Liason

Mr. J. M. McVaugh
Mr. R. K. Tu   Applied Mechanics
Mr. A. Bosco

Mr. G. L. Perrone - Aerodynamics

Mr. D. Edmonds - Design

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SMALL, HIGH-PRESSURE-RATIO COMPRESSOR MECHANICAL ACCEPTANCE TEST

SUMMARY

The Small, High-Pressure-Ratio Compressor Program was directed toward the analysis, design, fabrication and mechanical tests of a compressor for a pressure ratio of 6:1 and an airflow rate of 2.0 pounds per second under Contract NAS-3-14306.

The final report is in two volumes. The first volume (NASA CR-120941) covers the analysis, selection and design of the compressor and research package. The second volume (NASA CR-121193) covers the fabrication and mechanical testing of research package.

This volume presents a detailed description of the compressor rig test setup, and the various builds required to successfully demonstrate mechanical integrity of the design and the overspeed capability of the research package. Other mechanical tests consist of demonstrating adequate rotor dynamics, electrical isolation at the gas bearing trunnion-mounted diffuser and shroud, and the effect of operating parameter on impeller clearance of the compressor research package. This work was performed by AiResearch Manufacturing Company of Arizona, a Division of The Garrett Corporation, Phoenix, Arizona.

The compressor research package was mechanically checked out over a speed range of 20 to 120 percent of rated speed (80,000 rpm).

Instrumentation that was not a part of the original contract was installed to measure shroud-to-impeller tip clearance. These clearance measurements were aimed at determining the effects of rotor speed and pressure ratio on impeller-to-shroud clearance. This was done so that the final acceptance test could be run at radial and axial operating clearances from 0.009 to 0.011 inch.

Photographs of the compressor rig test setup are presented in figures 11a and 11b and three final assembly drawings of the test rig are included in Appendix III.

The mechanical checkout of the test rig, witnessed by NASA Project Manager, Mr. Robert Y. Wong, included the following mechanical tests:

1. An impeller overspeed test in a spin pit to 140 percent of design speed, to establish the mechanical overspeed safety margin of the rotor.

2. A turbo-compressor package acceptance test that included:

(a) A five (5) hour run at 100-percent design speed (80,000 rpm) to demonstrate the integrity of the package. This was accomplished with the NASA supplied turbine.

(b) A five (5) minute run at 120-percent design speed to demonstrate the short time overspeed capability of the turbo-compressor package.

3. The measurement of the variation in impeller to shroud clearance as affected by rotor speed and compressor pressure ratio was made so that proper setup clearance could be selected to give design clearance at design speed.

A third critical speed rotor dynamics problem was experienced at the upper end of the operating speed range of compressor rig Builds 1 and 2. A rotor dynamics test was therefore conducted using a "dummy" compressor having the same mass as the real rotor to determine a fix. As a result, the rig design was altered by modifying the bearing clearances to provide the hydraulic damping necessary to raise the discovered shaft bending mode out of the compressor operating speed range.

Carbon-face primary seals were checked regularly before each build with regard to achieving close runout and flatness tolerances. This action precluded any primary seal problems arising from the high rotor speed range of interest. However, frequent problems were encountered with the teflon secondary seals during testing. It was discovered that teflon seals were not flexible enough to perform the intended function while under the shaft and seal excursions experienced at high rotor speed. Seal leakage would occur whenever the compressor or turbine scavenge pressure was permitted to rise above -5 inch mercury gage (-2.47 psig). Finally, silicone O-ring seals were tried, replacing the teflon seals, and proved successful in this application.

A requirement of this contract was to provide a method to accurately measure compressor torque, and therefore compressor power absorption. Considerable effort was expended toward achieving a trunnion mounted diffuser-shroud-bellmouth assembly on hydrostatic gas bearings. This unique application of gas bearing technology provides a method of measuring aerodynamic reaction torque on the assembly, and therefore compressor impeller input torque. Problems experienced using this suspended assembly approach to accurately measure compressor torque were mainly torque measurement hysteresis and random assembly contact arising from eccentric loading or tilt of assembly.

The compressor research package successfully fulfilled the mechanical acceptance tests and was delivered to the NASA-Lewis Research Center on November 3, 1972.

1. INTRODUCTION

The Small, High-Pressure-Ratio Compressor Program was directed toward the analysis, design, fabrication and mechanical tests of a compressor for a pressure ratio of 6:1 and an airflow rate of 2.0 pounds per second under Contract NAS-3-14306.

This volume discusses the problems and modifications made to the mechanical hardware and compressor test rig assembly in order to perform the mechanical acceptance test. In the original contract of June 5, 1970, operation at design clearance for the acceptance test was not originally specified. A subsequent contract amendment was received on November 4, 1971, for determining the variation in impeller-to-shroud as a function at operating parameter (speed and pressure ratio) so that the acceptance test could be conducted at design clearance (0.009 to 0.011 inch). Instrumentation was installed in the impeller shroud to accomplish this objective.

The contracted requirements were performed through a series of six complete build assemblies, designated Builds 1 through 6, and several rig modifications requiring partial disassembly of these builds which were designated by alphabetical suffixes. The purpose of each of these builds is itemized below.

Build Number	Purpose
1	Trial assembly to determine fits and check stacking of parts.
1A	Completely instrumented assembly for first mechanical test using increased compressor axial clearances.
2	Second mechanical test with improved rotor runout for reducing high speed vibration.
2A	Build modification to improve hydraulic mount damping by oil supply orifice change.
2B	Dummy mass test to determine rotor critical speeds after inducer blade failure on Build 2A.
3	Capacitance probe clearance test and 5-minute overspeed test at 120-percent design speed using reworked inducer configuration.
4	Build to accomplish electrical isolation of the floating diffuser on the gas bearing and to operate at the design compressor clearances.
4A	Rebuild of Build 4 to correct seals which failed during test.

Build Number	Purpose
5	Rig assembly without oil seals to determine the feasibility of operation without the structural dampening of the front journal gas bearing support without experiencing a compressor wheel rub.
5A	Rig assembly with oil seals, front journal gas bearing support, modified compressor volute housing to achieve electrical isolation of the gas bearing, and a new inducer with radial hand finish work on the blades.
6	Build for 5-hour mechanical acceptance test with modified oil seal stacking and compressor face clearance.

2. DISCUSSION OF RESULTS

Build 1

The Inducer, SKP25657-1, Impeller, SKP25658-1, and Turbine Wheel, CR849373, were oversped on August 11, 1971. Two positions on each wheel were measured before and after overspeed. The following characteristics were observed from the tests:

P/N	Name	S/N	Before Dimension	After Dimension	Speed
SKP25657	Inducer	2	3.6505 3.6510	3.6506 3.6510	112,000 RPM
SKP25658	Impeller	101	5.3745 5.3745	5.3748 5.3748	112,000 RPM
CR849373	Turbine	1	4.1279 4.1274	4.1278 4.1274	112,400 RPM

Perceptable growth was observed on the impeller, SKP25658, only. Critical dimensions were taken prior to and after overspeed. Copies of the critical cards are shown in figures 1 through 3.

Several parts required rework prior to the first build. Inspection revealed that the diffuser assembly, SKP25710-1, required inside diameter remachining to clear the impeller tip diameter; the compressor shroud, SKP25668-1, had an incorrectly machined contour; the inlet bellmouth, SKP25708-1, was too short for accurate calibration and required a neck extension; and the inlet housing, SKP25666-1, was inadequately machined to achieve the required stacking dimensions. All hardware was corrected for the first build by October, 1971. Photographs of the impeller and inducer are shown in figures 4 through 8.

The pre-instrumentation build was completed to establish proper shimming and clearances. The gas bearing system was pressurized to assure that no binding existed. The gas bearings operated freely at 50 psig air pressure. The pressure was increased to 150 psig and no air hammering was evident. Photographs of the rotating group were taken and are shown in figures 9 and 10. The test rig was disassembled and the required hardware was sent to instrumentation.

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AiResearch Manufacturing Company of Arizona
QUALITY CONTROL
REINSPECTION RECORD

PART NUMBER CP 549373 C/L 1
PART NAME WHEEL, TURBINE

Next Assembly ________ C/L ____ Final Assembly ________ S/N 1

NO.	Dimension and Location		BP max./min.	Before	After	Remark
1	CURVIC O.D.		2.000 / ±.001	2.000	2.000	
2	OVERALL LENGTH		1.564 / 1.562	1.5637	1.564	
3	NOTE 12		.0004 / [illegible]	.0002		
4	I.D.		.3757 / .3754	.3757 / .3756	.3757 / .3756	
5	NOTE 1		.0002 / FIR	.0002		
6	BALANCE VERIFICATION					
7						
8						
9						
10						

Inspection Before [illegible] Date 8-7-71 Quality Control ________ Date ____
After [illegible] Date 8/30/72 Engineering ________ Date ____

Figure 1.

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P5467

AiResearch Manufacturing Company of Arizona
QUALITY CONTROL REINSPECTION RECORD

PART NUMBER 3KP25658 C/L —
PART NAME IMPELLER

Next Assembly ______ C/L ___ Final Assembly ______ S/N 101

NO.	Dimension and Location		BP max./min	Before	After	Remark
1	BORE, LAB. SIDE	2D	.7901 / .7885	.7888	7889	
2	BORE, INDUCER SIDE	4D	.7005 / .7000	.7002	7004 / 7003	
3	⊥ OF 'A' TO C-B	3E	.0005	.0001	.0002	
4	LABRYNTH O.D.	3G	2.005 / 2.004	2.005	2.005	
5	⊥ C-B	4C	.0002	.0001	.0001	
6	INTERNAL SPLINE	8E	.5765 MAX	.5770	.5713	BETWEEN .0864 PINS
7		8E	.0004 Max Ref	OK TO GAGE	OK TO [illegible]	TOOTH 11 T-209270
8		3E	.0015	.001	.0008	PITCH DIA W/ B-C DIA.
9	COMPARATOR CHART	6C		WAIVED PER ENG. USED PARAGONS CHARTS		L687507
10	L & Z COORDINATES	6B		"		L687501 "

Inspection Before [signature] Date 7-19-71 Quality Control ______ Date ______
After [signature] Date 8-17-71 Engineering ______ Date ______

Figure 2.

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
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P5467

AiResearch Manufacturing Company of Arizona
QUALITY CONTROL
REINSPECTION RECORD

PART NUMBER SKP25657-[illegible] C/L
PART NAME INDUCER, IMPELLER

Next Assembly SKP25711-1 C/L B Final Assembly SKP25775-1 S/N 2

NO.	Dimension and Location		BP max./min	Before	After	Remark
1	-A- DIA	E5	.7000 / .6998	.700	.700	
2	-B- SURFACE	F5	⊥ A / .0003	.0009	.0009	
3	-C- DIA	E6	1.403 / 1.402	1.403 / 1.402	1.403 / 1.402	
4	-C- DIA	E6	↗ A-B / .0005	.0003	.0003	
5	ID	D5	.4043 / .4040	.4043	.4040	
6	.4043-.4040 ID	D5	↗ A-B / .0005	.0003	.0004	
7	END FACE	C6	⊥ C / .0005	.0002	.0002	
8	RECESSED FACE	C6	⊥ A / .0005	.0003	.0004	LESS BALANCE AREA.
9	TABLET ~ COMPARATOR					BASE SECT. MID SECT.
10	CHART ATTACHED					TIP SECT.

Inspection Before [illegible] Date 8/27/71 Quality Control ____ Date ____
After [illegible] Date [illegible] Engineering ____ Date ____

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Figure 3.

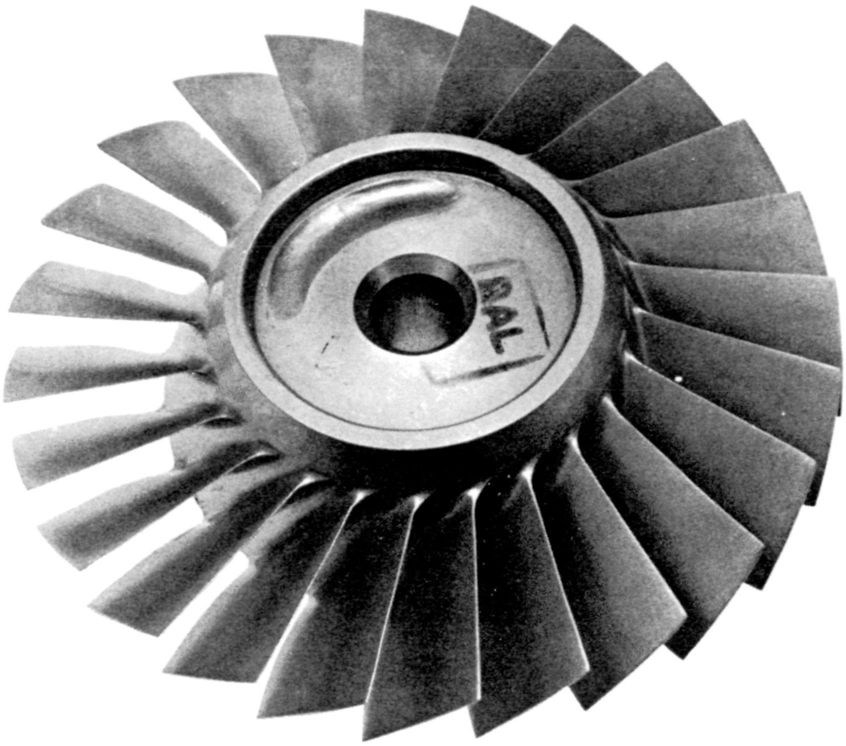

Figure 4. Front View of Inducer SKP25657-1.

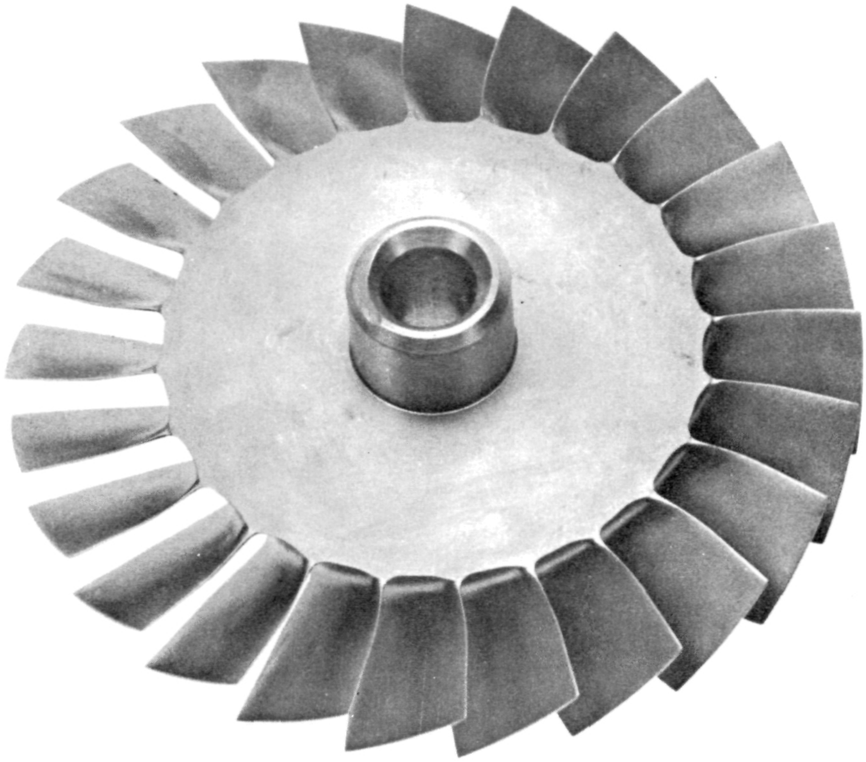

Figure 5. Back View of Inducer SKP25657-1.

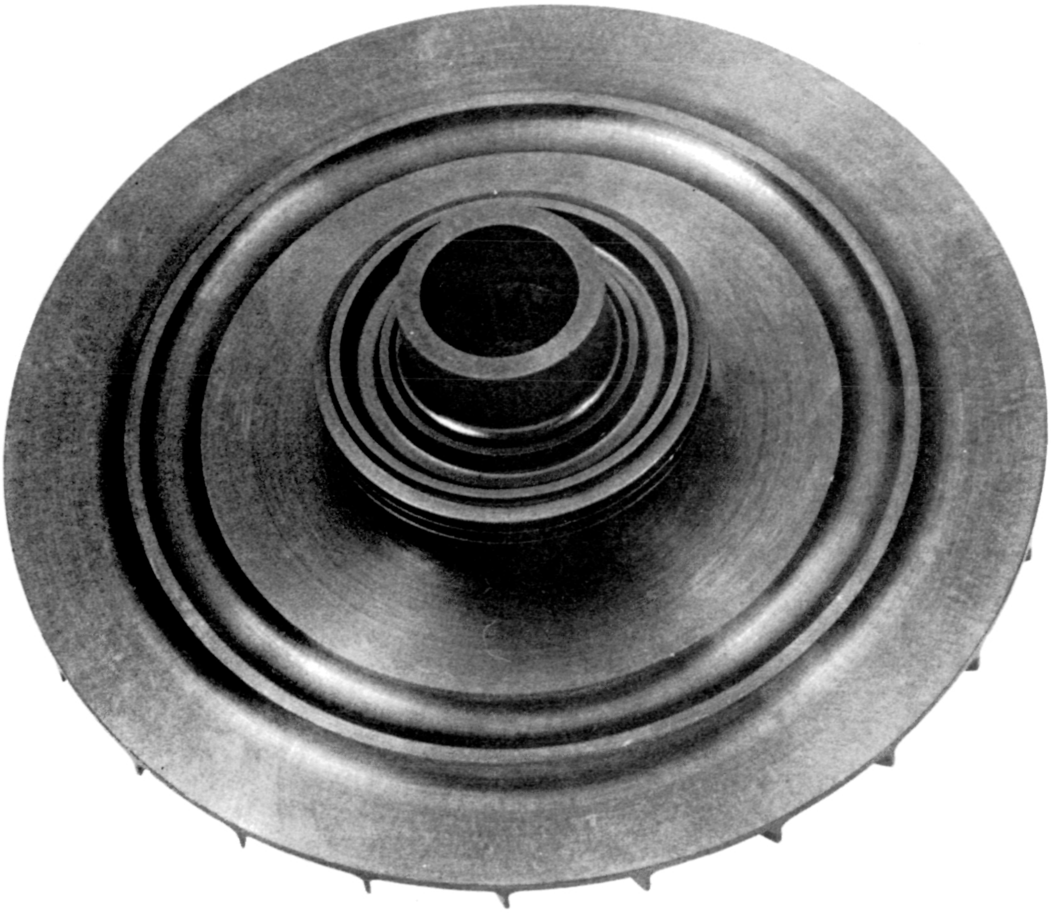

Figure 6. Back View of Impeller SKP25658-1.

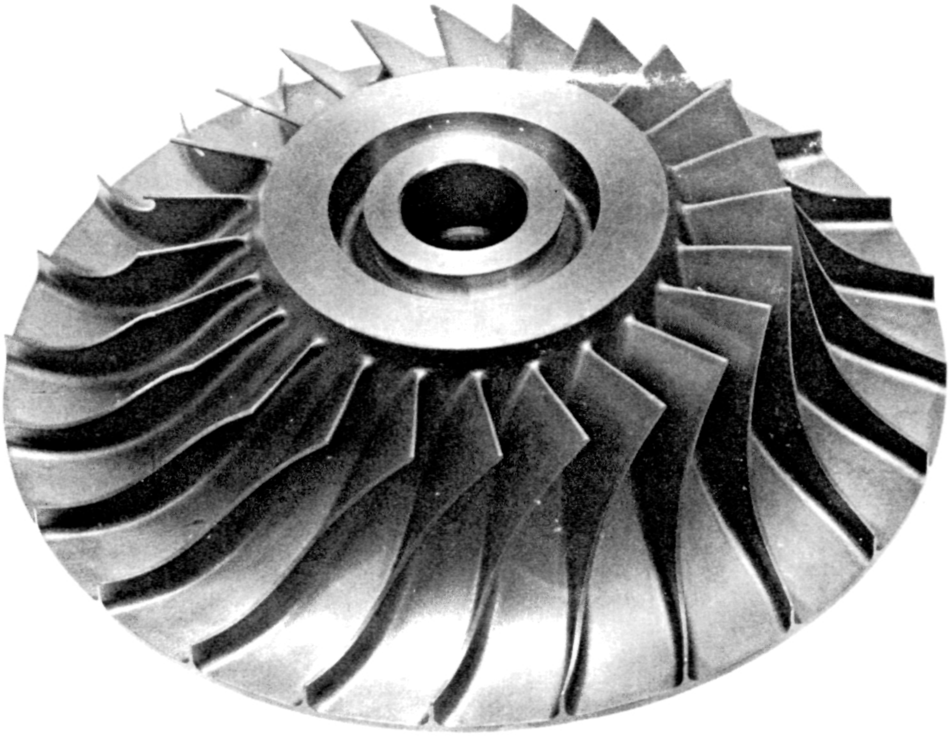

Figure 7. Front View of Impeller SKP25658-1.

Figure 8. Inducer and Impeller Assembly.

Figure 9. View of Rotating Group (Turbine End).

MP-31422

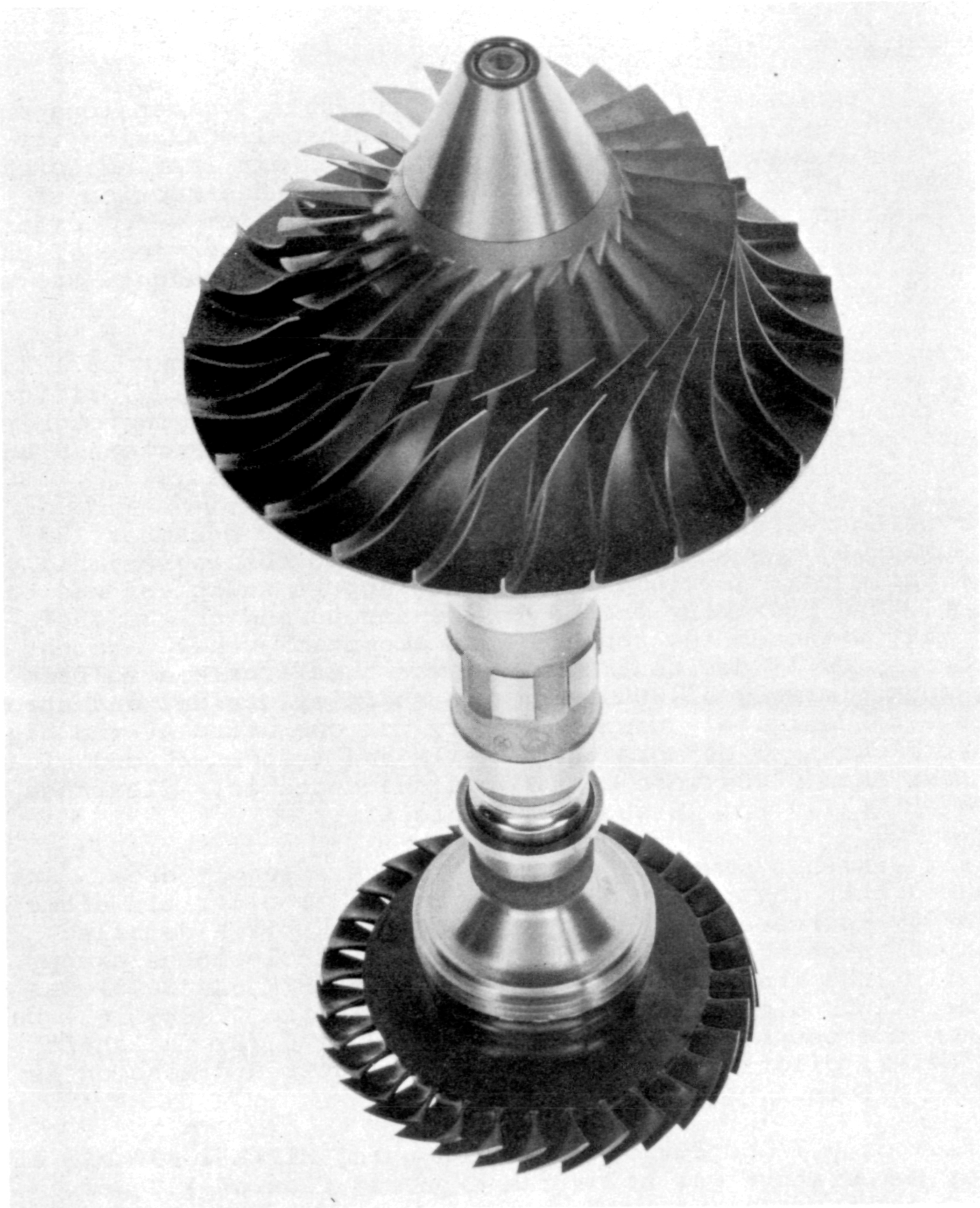

Figure 10. View of Test Rotating Group (Compressor End).

Builds 1A and 1B

The Build 1A test rig was assembled per Table I dimensions on 11 December 1971 and is shown installed in test cell CA2 with its supporting lubrication system as figure 11. The air bearing assembly exhibited a 15 to 30 in.-lb hysteresis from one direction of torque application to the other. Speeds to 72,000 rpm were performed and a critical speed problem was identified at the upper end of the speed range. Lissajous photos of the Bentley probe readouts are shown as figures 12 and 13.

The rig was removed from the test cell on 16 December 1971 and returned to the assembly area. Several of the gas bearing orifices were found to be blocked. The parts were disassembled, thoroughly cleaned, and reassembled. The test cell air system was cleaned and larger capacity air filters (10 microns) were installed.

After being reinstalled in the test cell on 20 December 1971, the rig, Build 1B, was accelerated to 16,000 rpm for approximately 60 minutes while test instrumentation and support equipment was being checked out. The proximity probes in the shroud and on the shaft indicated that although the runouts were acceptable they were not as closely controlled as desired. The rig was then accelerated from 16,000 to approximately 32,000 rpm. The shaft excursions and shroud clearances were definitely unsatisfactory for operation at the higher speed shaft excursions of more than 2 mils and inducer-shroud clearances of less than 0.002 were considered unacceptable. Clearance data taken at 16,740 rpm is shown in Table II.

The rig was shut down and returned to the assembly area. Inspection revealed that the runout dimension on the impeller tip diameter (P/N SKP25658) increased from 0.001 to 0.0025 inch. A detailed inspection of parts showed that all were within tolerances except for two: (1) the inner pilot on the impeller (P/N SKP25658) was found to be within dimensional tolerance (0.7901 to 0.7885 in.) but tapered from the outer point to the inner end, and (2) the shaft (P/N SKP25641-1) pilot O.D. was 0.7873 in. (required dimension is 0.7879 to 0.7876 in.).

Oil flow of 0.7 gpm during operation using MIL-L-23699 oil at 150°F inlet temperature was below the expected flow of 1.2 gpm.

It was concluded that before attempting further testing of the rig, the following corrective action should be taken: (1) close the tolerances on the impeller shaft pilot, and (2) increase the oil flow to assure a sufficient supply to the hydraulic bearing mounts.

TABLE I.

BUILD 1A DATA SHEET
NASA 6:1 COMPRESSOR

A. Runout

1. Turbine

a.	O.D.	0.0003
b.	Front Face	0.0003
c.	Knife Edge Seal	0.0002

2. Compressor

a.	O.D.	0.001
b.	Back Face	0.002
c.	Knife Edge Seal	0.0003

B. Balance	Max Allowed	Actual
1. Turbine	0.017 Oz-In.	0.0112
2. Compressor	0.023 Oz.-In.	0.0208

C. Clearances	B/P	Actual
1. Turbine Bearing Housing to Seal Retainer	0.023-0.027	0.025
2. Compressor Bearing Housing to Seal Retainer	0.001-0.003	0.001
3. Diffuser Knives to Discharge, Turbine End	0.003-0.005	
4. Diffuser Knives to Discharge, Compressor End	0.003-0.005	
5. Turbine Wheel Clearance	0.023-0.027	
6. Compressor Face Clearance	0.021-0.023	

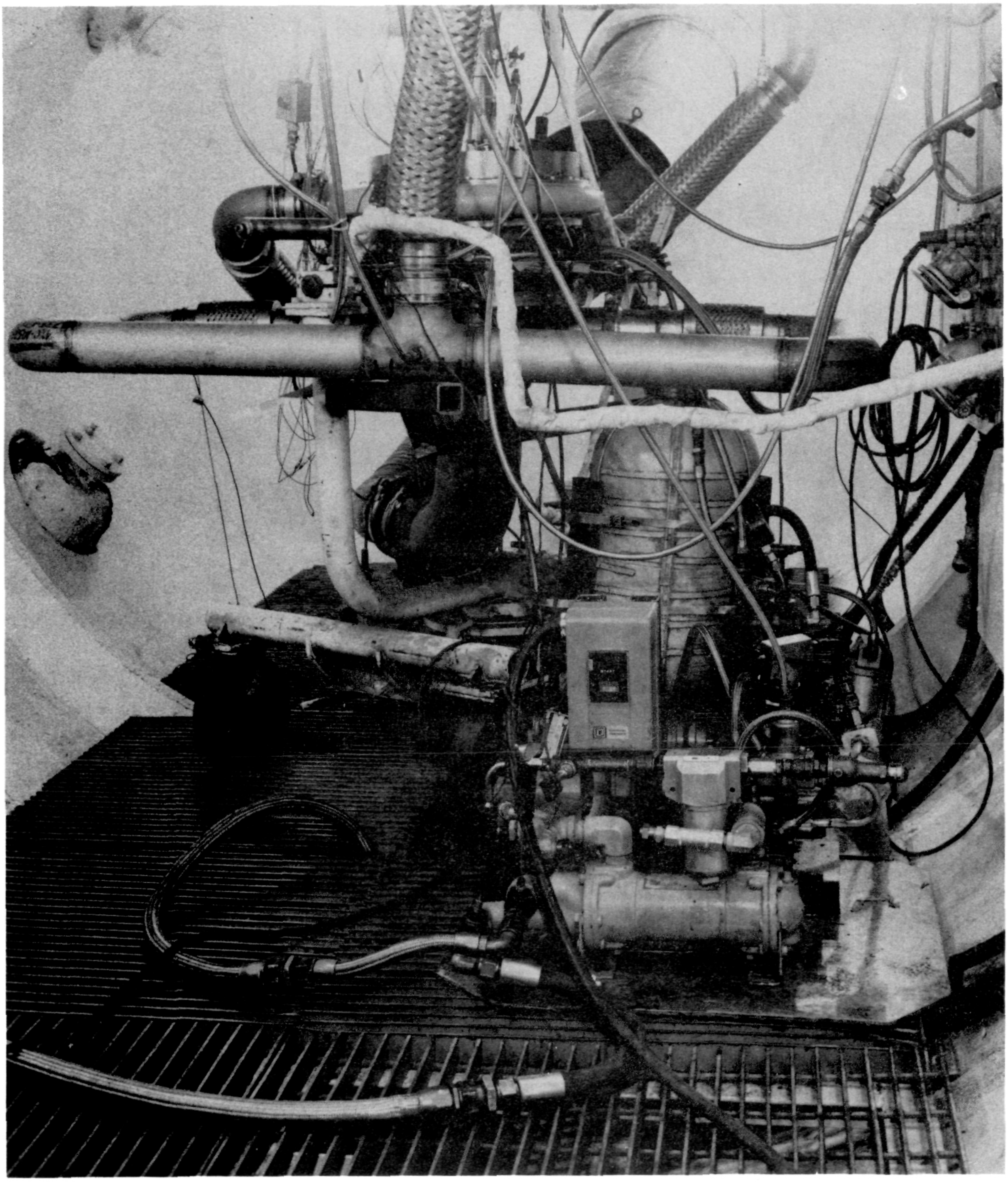

Figure 11a. Compressor Rig Test Setup With
Supporting Lubrication System.

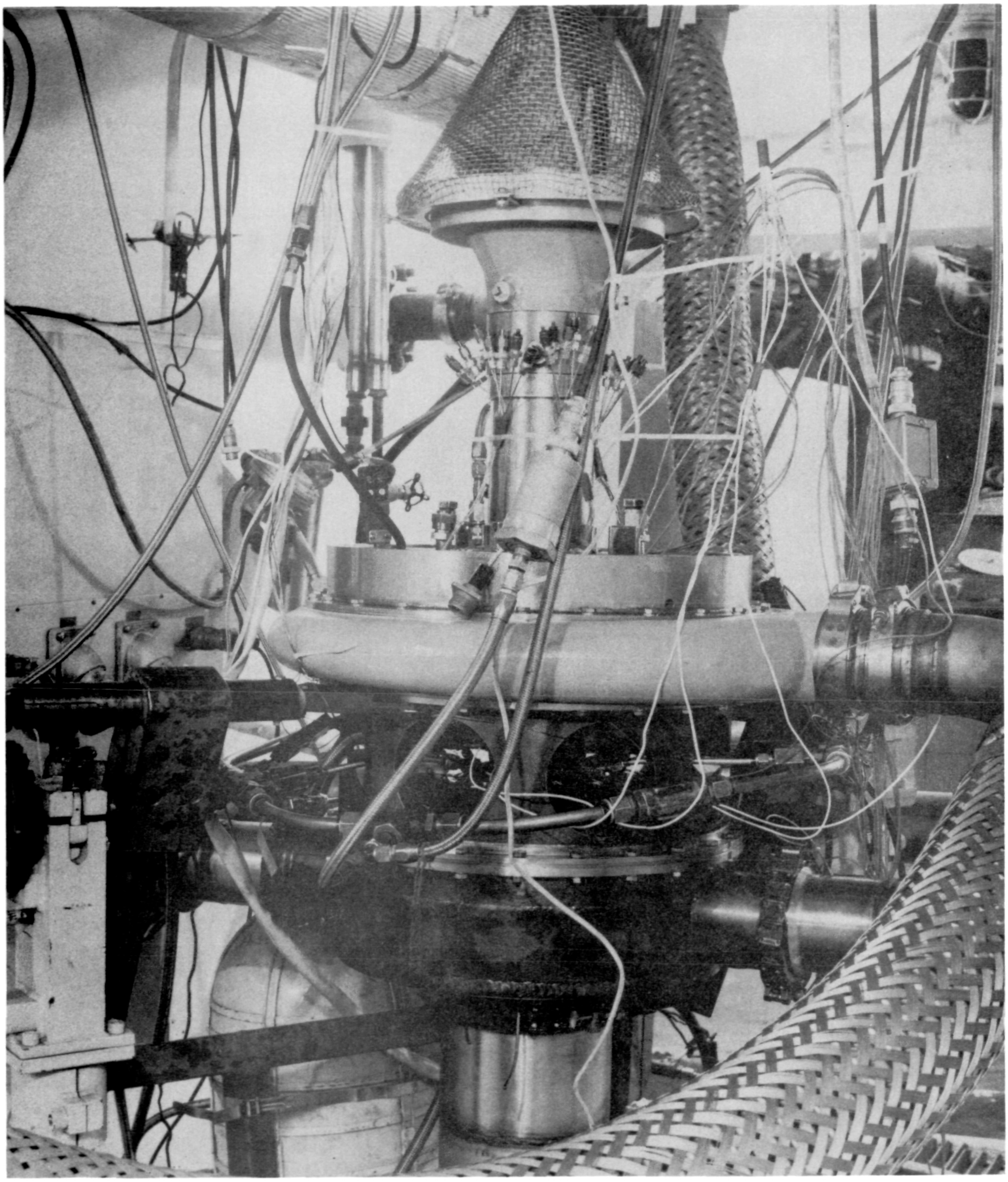

Figure 11b. Close-up of Compressor Research Package.

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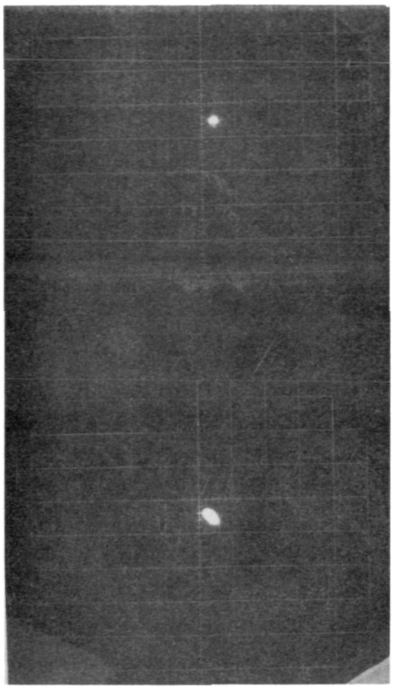
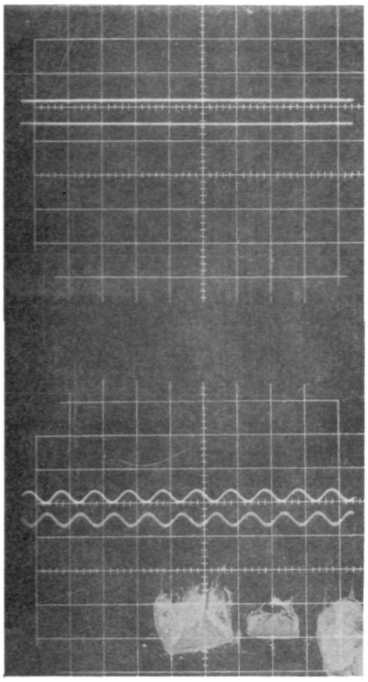
NASA 6:1 COMPRESSOR RIG; BUILD 1A LISSAJOUS TRACES

COMPRESSOR
END

TURBINE
END



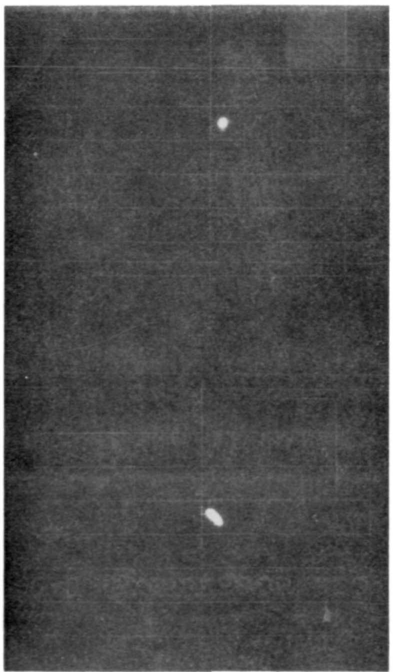
UNIT NOT ROTATING



UNIT NOT ROTATING
X = 0.2 MILLISEC/DIV.
Y = 1.25 MILS/DIV.

COMPRESSOR
END

TURBINE
END



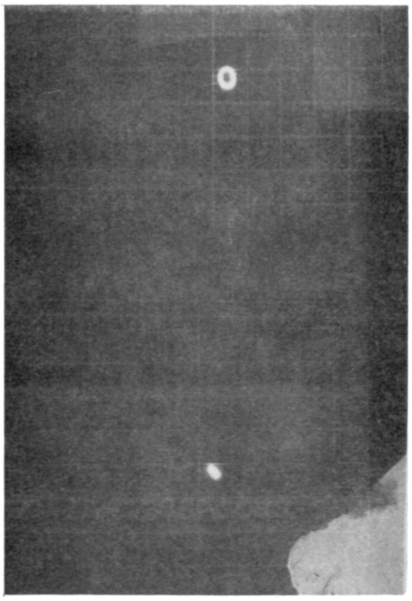


Figure 12.

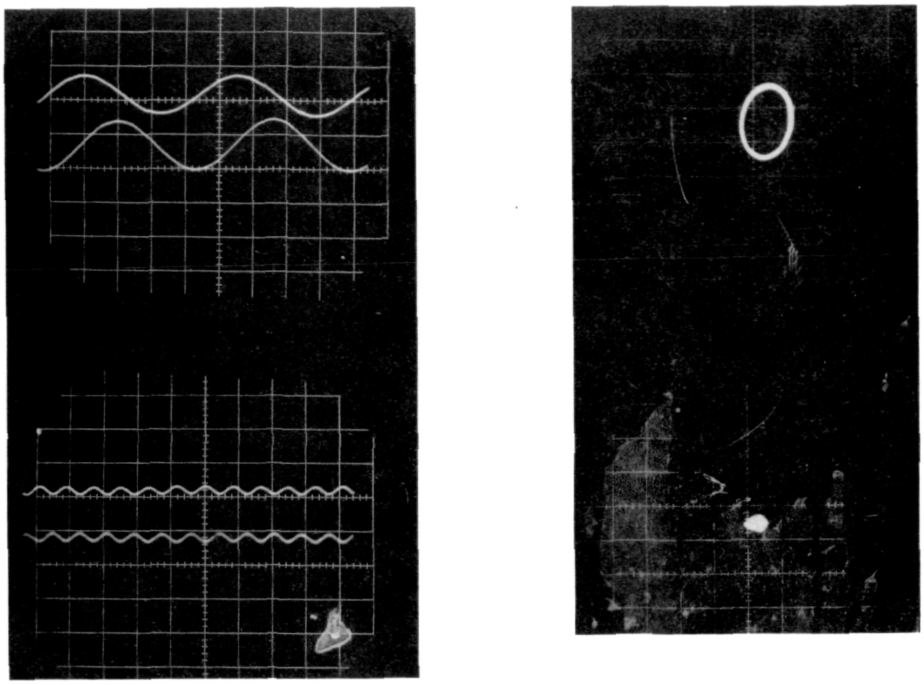
32,000 RPM

48,000 RPM
LISSAJOUS SCALES:
1.25 MIL/DIV;, 0.25 VOLTS/DIV.
TEST DATE: 16 DECEMBER 1971

NASA 6:1 COMPRESSOR RIG BUILD 1A LISSAJOUS TRACES

COMPRESSOR
END

TURBINE
END



64,000 RPM

72,000 RPM

X = 0.2 MILLISEC/DIV.
Y = 1.25 MILS/DIV.

LISSAJOUS SCALES:

1.25 MILS/DIV.
0.25 VOLTS/DIV.

TEST DATE:
16 DECEMBER 1971

Figure 13.

TABLE II.

BUILD 1B CAPACITANCE PROBE CLEARANCE DATA

Probe Number		Millivolt Reading	Shroud to Wheel Clearance, inches
1	Radial Probes	0.128	0.006
2		0.127	0.0038
3		0.128	0.0055
4		0.128	0.004
9	Axial Probes	0.031	0.054
10		0.032	0.054
11		0.031	0.051
12		0.031	0.055

The shaft pilots were plated with electroless nickel to close the tolerances on the impeller and then ground to a closer tolerance (0.7880 to 0.7885 in.). In addition, the shafts (P/N SKP 25641) were hard chrome plated over the pilots and reground to a closer tolerance (0.7877 to 0.7879 in.). To increase oil flow the test cell oil system was modified to allow oil temperature above 200°F and the oil was changed to MIL-L-7808.

A precautionary measure must be observed during assembly and disassembly of the shaft, SKP25641-1, and compressor bearing to avoid breakage of the alignment pin in the turbine end seal retainer SKP25647-1. Tools T-211574 and T-211556 are provided for bearing assembly to prevent load application to this pin.

Builds 2, 2A, and 2B

On January 7, Build 2 test rig was installed in test cell CA-2 without the compressor shroud as a precautionary measure for mechanical checkout of the bearing system. Build dimensions are shown in Table III. The diameter of the impeller bore had been reduced by nickel plating and the shaft was hard-chrome plated. The rig was accelerated to 75,000 rpm but the Bently proximity probes indicated that shaft orbiting was not properly controlled. Test data taken appears in Table IV. Lissajous traces are shown in figure 14.

The rig was removed from the test cell and disassembled. Examination of the parts indicated that there was no damage. To dampen shaft orbiting the diameter of the hydraulic-mount supply orifice was increased from 0.036 to 0.050 inch to supply a larger quantity of oil. In addition, the mechanical pinning of the bearings was eliminated to prevent the possibility of binding.

On January 21, Build 2A rig was installed in the test cell. The shaft precessed at speeds above 30,000 rpm. The rig reached the design speed of 80,000 rpm but during rolldown a large excursion accompanied by erratic movement was observed on the oscilloscope at approximately 56,000 rpm. Test data taken appears in Table V. Lissajous traces shown in figures 15 through 17 indicate the shaft orbit was still not acceptable.

Examination of the rig revealed that two adjacent blades on the SKP25657-1 inducer were missing. An investigation was undertaken to determine the cause of inducer failure. The damaged inducer and other damaged hardware are shown in figures 18 through 24.

Robert Wong (NASA) visited AiResearch on January 27 to inspect the failed hardware and discuss the progress of the investigation. Another meeting was held on March 1 and 2, 1972 with NASA-Lewis personnel Warner Stewart, Hal Rohlik, and Bob Wong to discuss the inducer blade failure. The failure analysis is presented in Appendix I.

TABLE III.

BUILD 2 DATA SHEET
NASA 6:1 COMPRESSOR

A. Runout

1. Turbine

a.	O.D.	0.0003
b.	Front Face	N/A
c.	Knife Edge Seal	0.0002

2. Compressor

a.	O.D.	0.001
b.	Back Face	0.0012
c.	Knife Edge Seal	0.0005

B. Balance

Balance	Max Allowed	Actual
1. Turbine	0.017 Oz-In.	0.0026
2. Compressor	0.023 Oz.-In.	0.0128

C. Clearances

Clearances	B/P	Actual
1. Turbine Bearing Housing to Seal Retainer	0.023-0.027	
2. Compressor Bearing Housing to Seal Retainer	0.001-0.003	
3. Diffuser Knives to Discharge, Turbine End	0.003-0.005	
4. Diffuser Knives to Discharge, Compressor End	0.003-0.005	
5. Turbine Wheel Clearance	0.023-0.027	
6. Compressor Face Clearance	0.021-0.023	

FORM F-798

NASA 6:1 COMPRESSOR RIG

DATE: 7 June 72
OPERATOR: O Bennett
ASSISTANT:

Speed	0		16,000	32,000	32,000		48,000		64,000	75,000	80,000	*	96,000			
Oil Inlet Pressure PSIG			47.0	82.0	66.0		74.0		75.0							
Oil Inlet Temperature °F			184	204	200		210		208							
Oil Flow GPM / CPS			.51	181/	181/		196/		203/							
Compressor Bearing Temperature °F					/											
#1			175	—	180		—		—							
#2			—	—	—		—		—							
#3			—	—	—		—		—							
Turbine Bearing Temperature °F																
#1			192	203	198		240		252							
#2			192	203	198		240		252							
#3			192	203	198		240		261							
Thrust			65.0	45	90		55		100							
#1												✓				
#2																
#3																
Thrust Chamber Pressure PSIG			13.5	0	16.5		0		0			✓				
Vibration (Diff) g's																
Vibration (Housing) g's			0	0	0		0		3.5							
Shaft Excursion																
#1 Turbine																
#2 Turbine																
#1 Compressor																
#2 Compressor																
Turbine Inlet Temperature °F			55	204	55		235		290			✓				
Turbine Inlet Pressure PSIG				10.0			24.0		48.0							
Turbine Discharge Pressure PSIG												✓				
Diffuser Force Lbs												✓				
Gas Bearing Pressure (Top) PSIG																
Gas Bearing Pressure (Btm) PSIG																

* DO NOT EXCEED 5 MINUTES TOTAL OPERATION AT THIS SPEED. (✓ items Only)

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

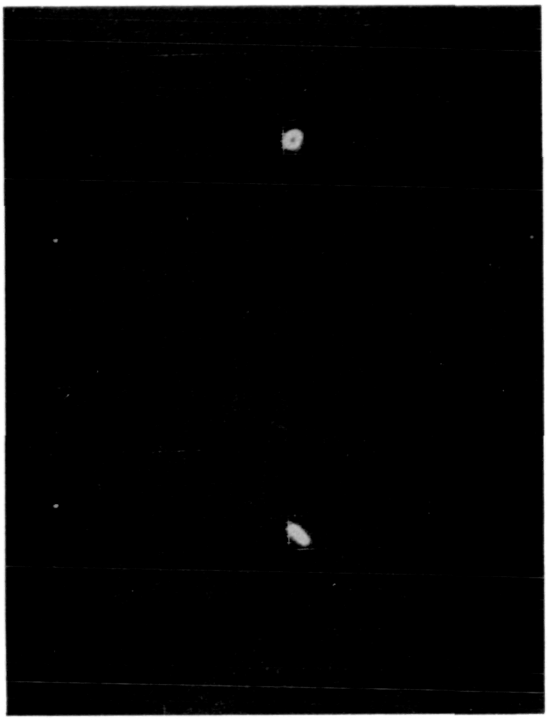
NASA 6:1 RIG BUILD 2

AiResearch Manufacturing Company of Arizona

TABLE IV.

NASA 6:1 COMPRESSOR RIG
BUILD 2 LISSAJOUS TRACE

COMPRESSOR
END



TURBINE
END

16,800 RPM

LISSAJOUS SCALES: 1.25 MILS/DIV.
0.25 VOLTS/DIV.

TEST DATE: 7 JANUARY 1972

Figure 14.

NASA 6:1 COMPRESSOR RIG

DATE: 1-21-72
OPERATOR: Bennett
ASSISTANT:

Speed	0	32000	60140	63040	7,920
Oil Inlet Pressure PSIG		70.0	70.0	70.0	70.0
Oil Inlet Temperature °F		91	194	196	220
Oil Flow GPM		.39	1.25	1.25	.10
Compressor Bearing Temperature °F					
#1		90	203	207	218
#2		90	203	207	210
#3		90	203	207	210
Turbine Bearing Temperature °F					
#1		101	215	218	227
#2		75	—	—	—
#3		101	215	218	225
Thrust					
#1					
#2					
#3					
Thrust Chamber Pressure PSIG			155	170	160
Vibration (Diff) g's					
Vibration (Housing) g's					
Shaft Excursion					
#1 Turbine					
#2 Turbine					
#1 Compressor					
#2 Compressor					
Turbine Inlet Temperature °F		65	45	50	130
Turbine Inlet Pressure PSIG		12.0	40	50	105
Turbine Discharge Pressure PSIG		0	0	0	—
Diffuser Force Lbs					
Gas Bearing Pressure (Top) PSIG					
Gas Bearing Pressure (Btm) PSIG					

ALTITUDE EQUIPMENT DIVISION
CALCULATED
RECORDED
DRAWN
CHECKED
APPROVED

NASA 6:1 COMPRESSOR RIG
BUILD 2A DATA

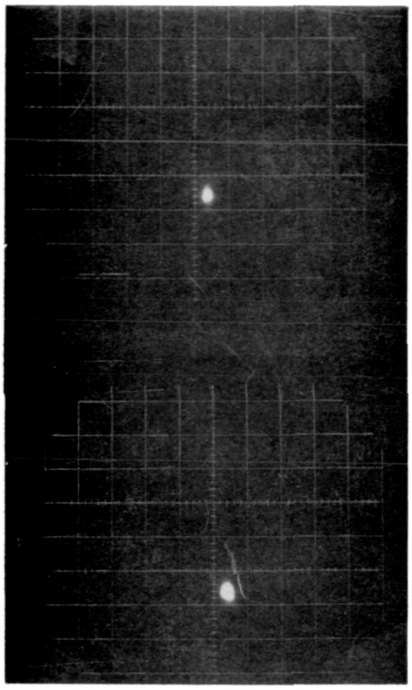
TABLE V.

AiResearch Manufacturing Company of Arizona

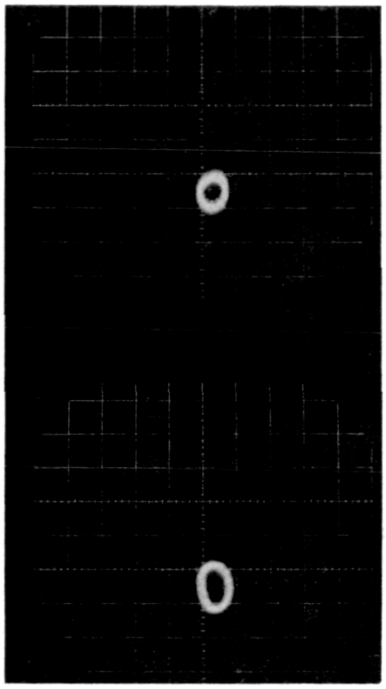
NASA 6:1 COMPRESSOR RIG
BUILD 2A LISSAJOUS TRACES

COMPRESSOR END

TURBINE END



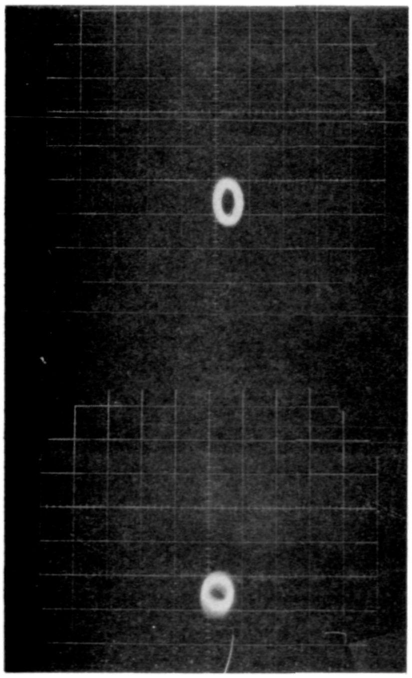
16,900 RPM



32,000 RPM

COMPRESSOR END

TURBINE END



40,000 RPM

APPROXIMATE SCALE: 1.0 MILS/DIV.
0.2 VOLTS/DIV.

TEST DATE: 21 JANUARY 1972

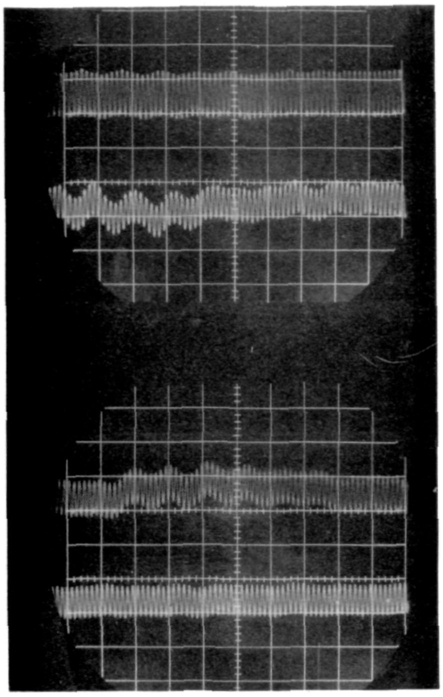
Figure 15.

NASA 6:1 COMPRESSOR RIG
BUILD 2A SHAFT EXCURSION TRACES

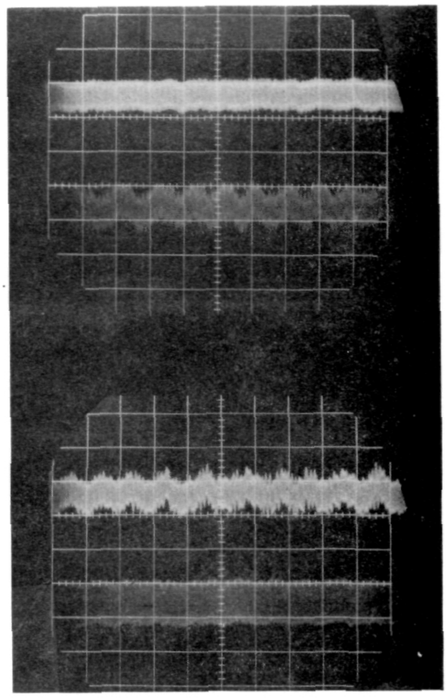
x AXIS = TIME, y AXIS = AMPLITUDE

COMPRESSOR END

TURBINE END



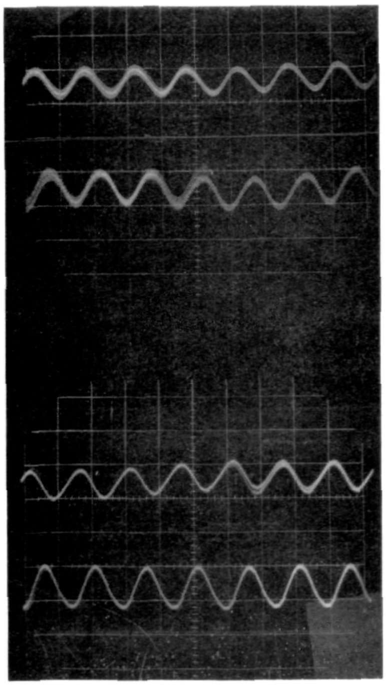
39,860 RPM
x = 0.01 SEC/DIV.
y = 0.001 INCH/DIV.



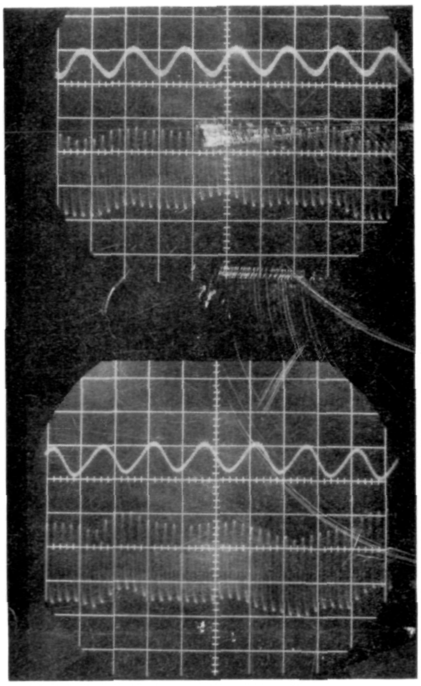
39,860 RPM
x = 0.1 SEC/DIV.
y = 0.001 INCH/DIV.

COMPRESSOR END

TURBINE END



40,000 RPM
x = 0.001 SEC/DIV.
y = 0.001 INCH/DIV.



40,000 RPM
x = 0.001 SEC/DIV.
y = 0.001 INCH/DIV.

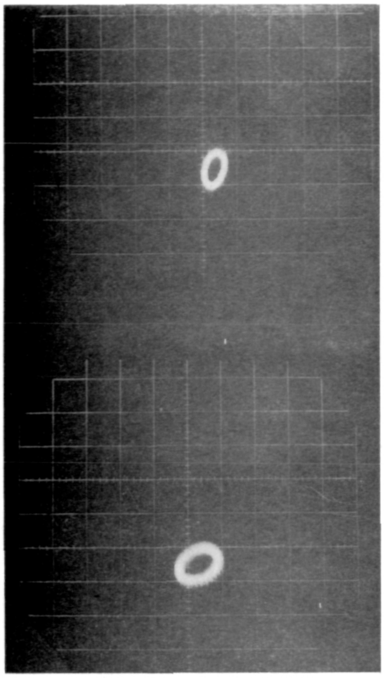
Figure 16.

TEST DATE: 21 JANUARY 1972

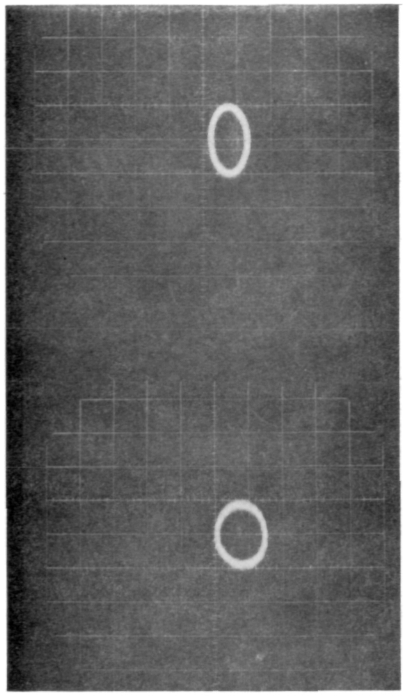
NASA 6:1 COMPRESSOR RIG
BUILD 2A LISSAJOUS TRACES

COMPRESSOR END

TURBINE END



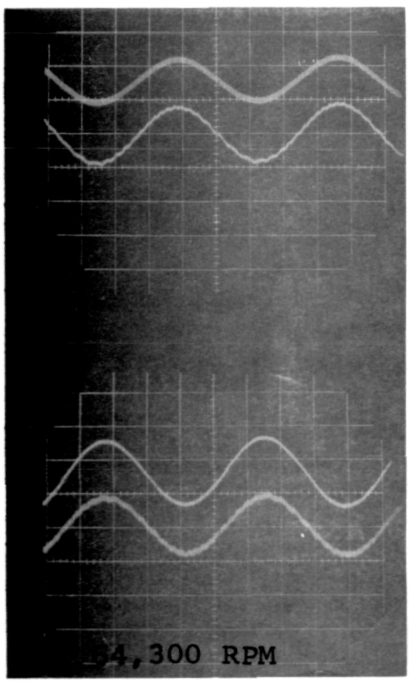
50,600 RPM



63,800 RPM

COMPRESSOR END

TURBINE END



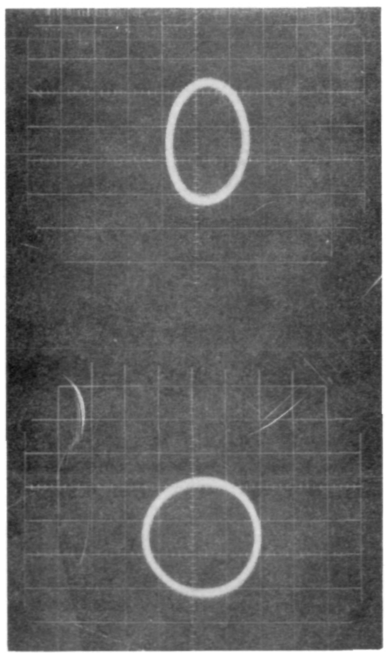


Figure 17.

77,500 RPM

SCALES: 1 MIL/DIV.
0.2 VOLTS/DIV

TEST DATE: 21 JANUARY 1972

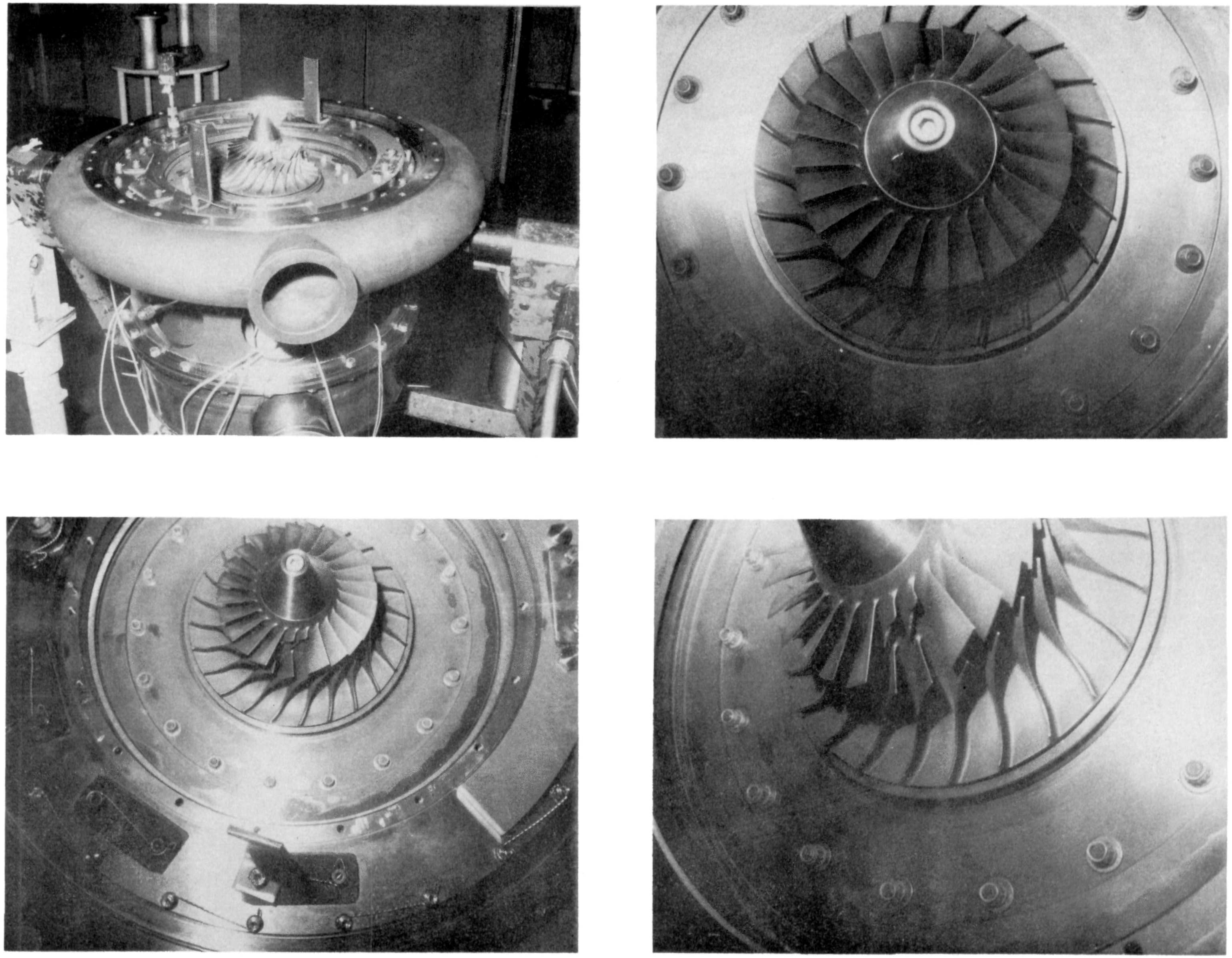


Figure 18. NASA 6:1 Compressor Test Rig
SKP25657-1 Inducer Failure.

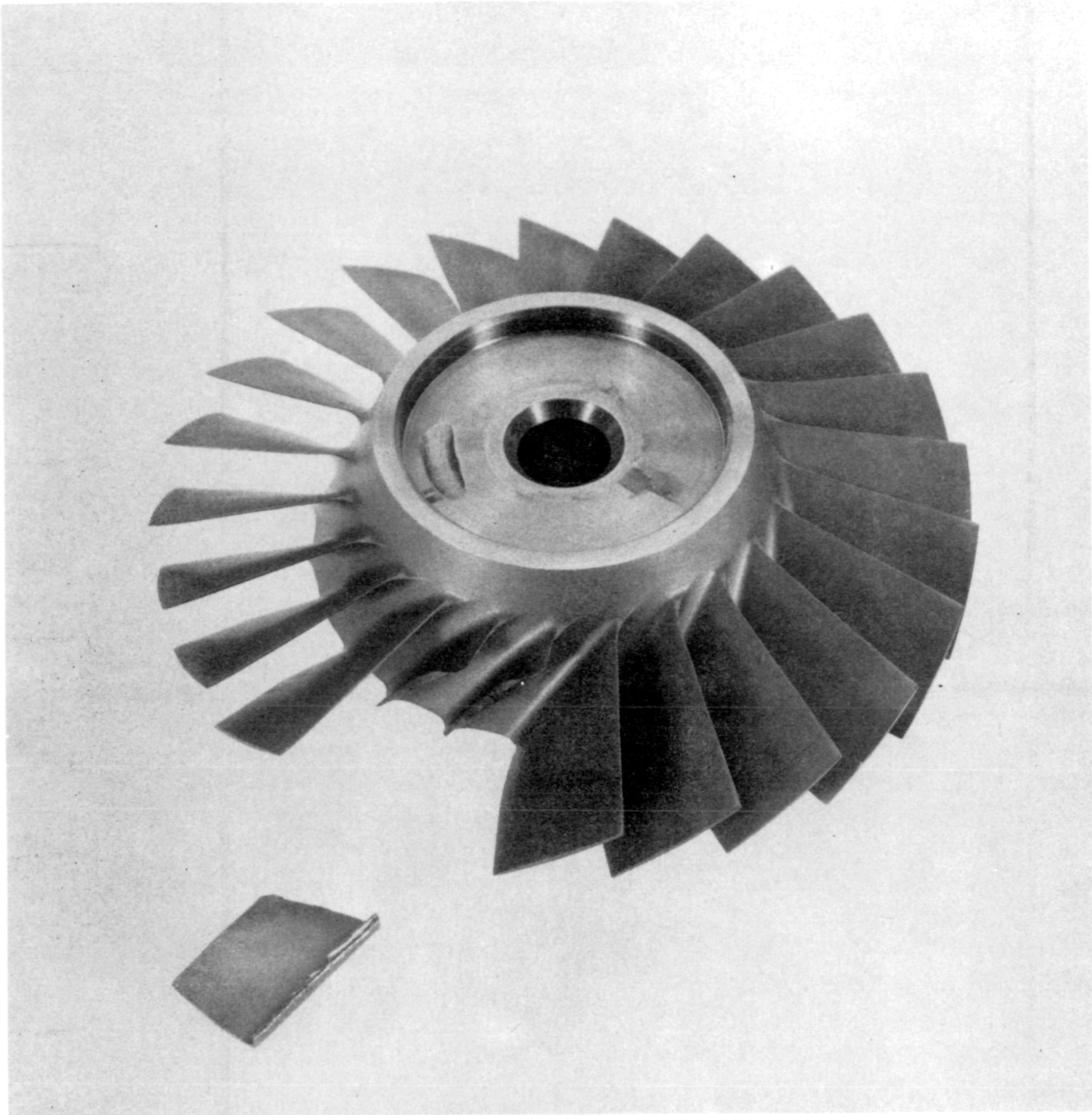


Figure 19. NASA 6:1 Compressor Test Rig SKP25657-1 Inducer Failure.

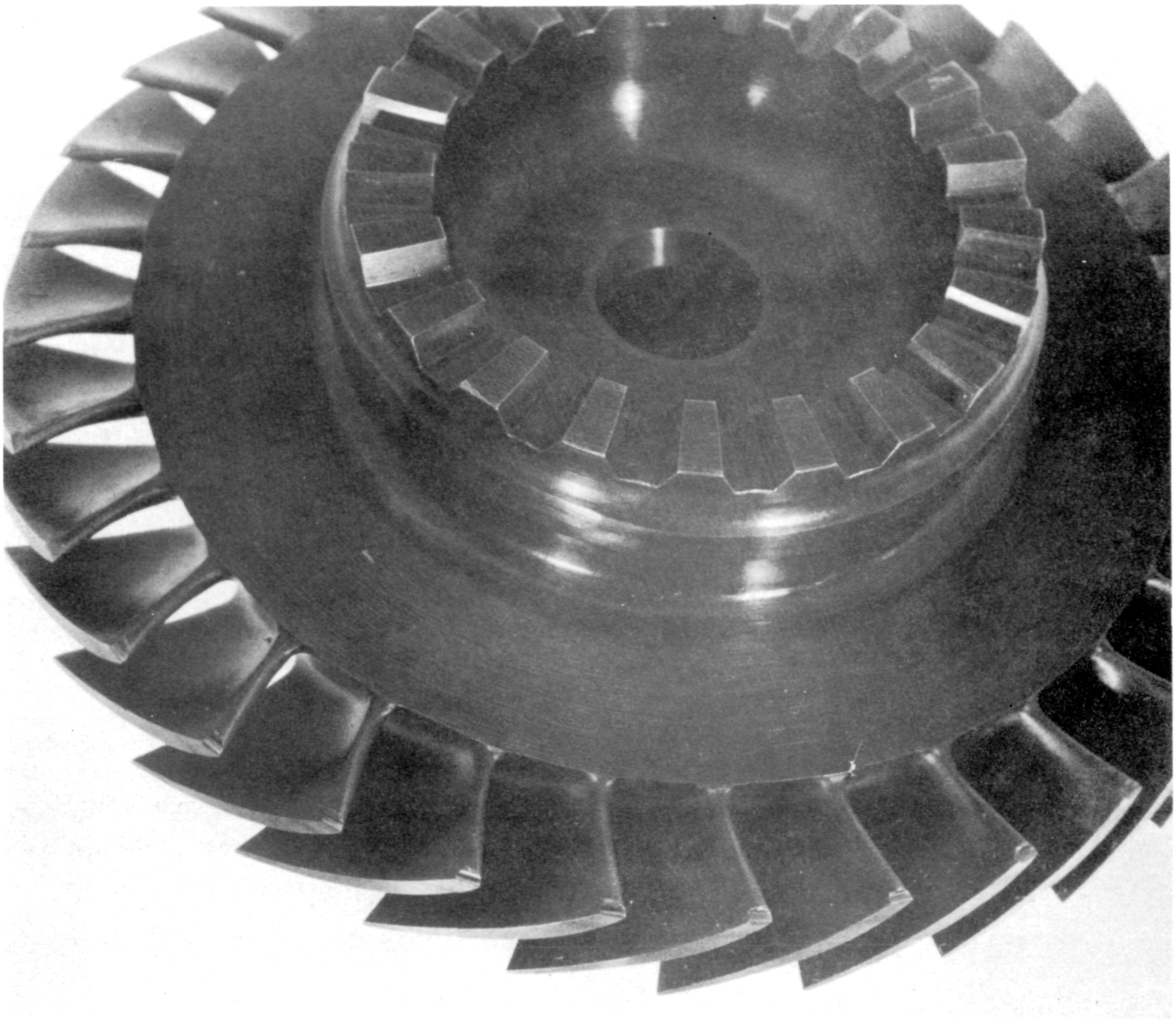


Figure 20. NASA 6:1 Compressor Test Rig, Inducer Failure, NASA Turbine Wheel Rub.

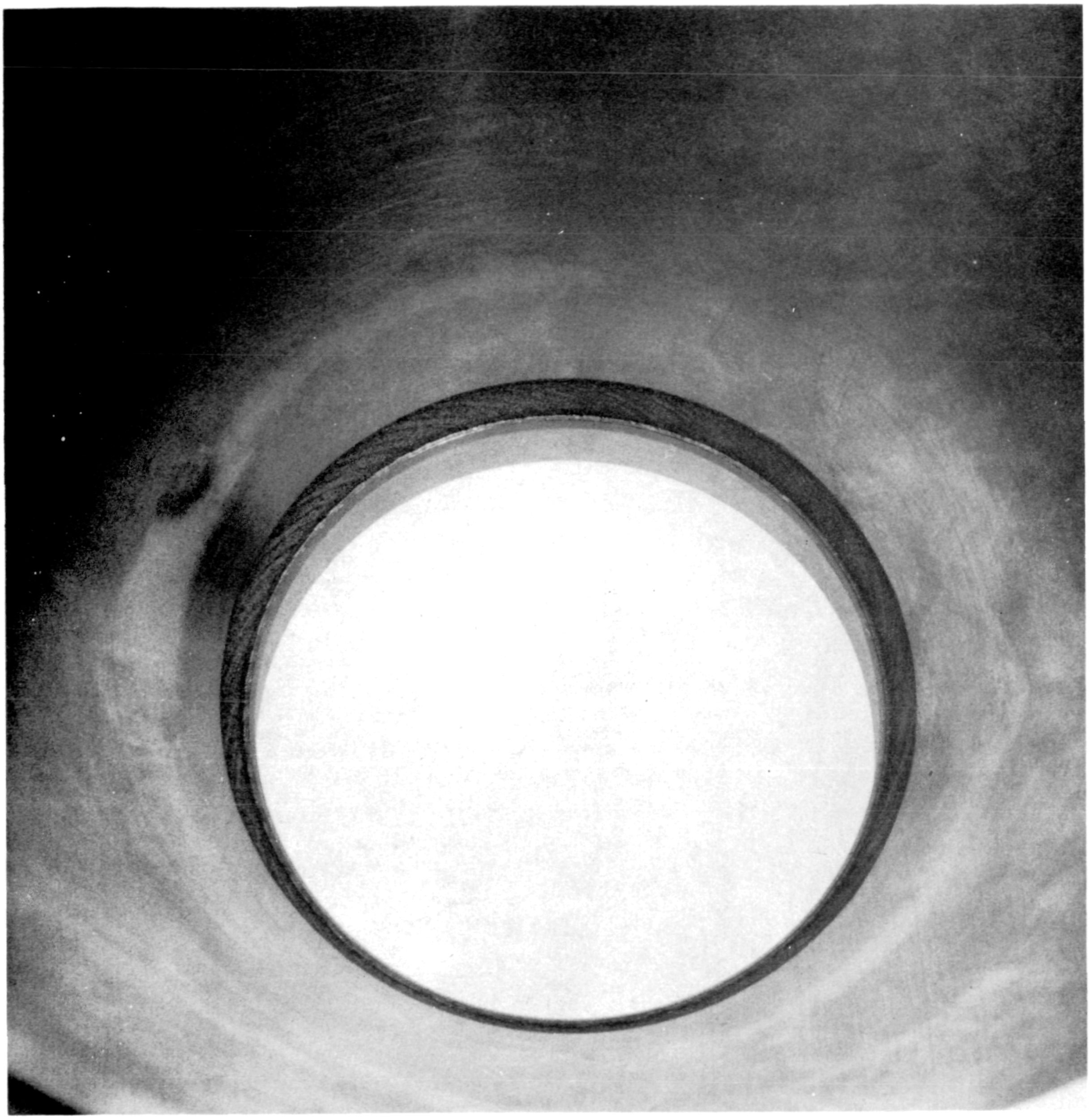


Figure 21. NASA 6:1 Compressor Test Rig, Inducer Failure, NASA Turbine Exhaust Duct Abradable Coating Wear.

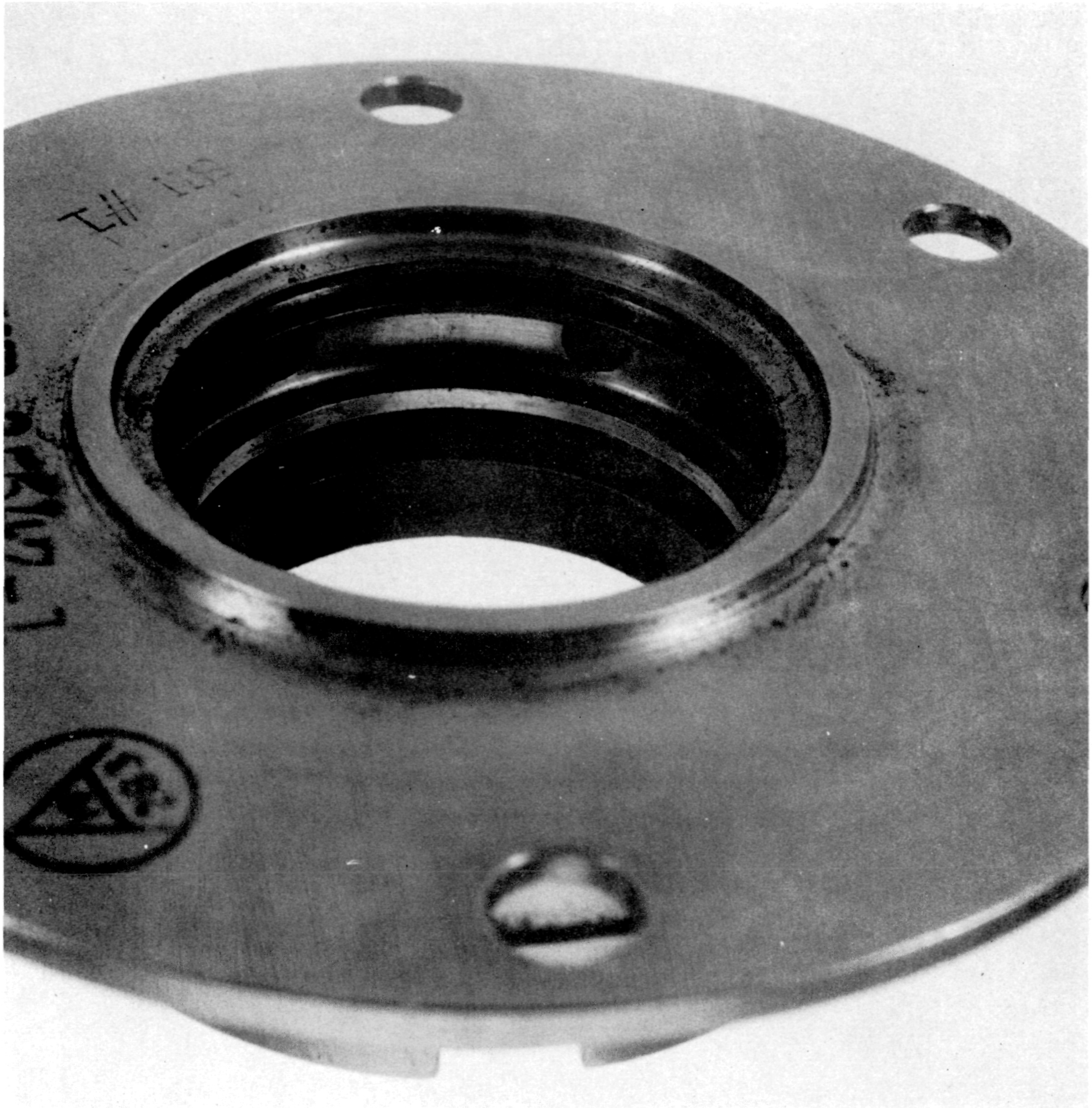


Figure 22. NASA 6:1 Compressor Test Rig, Inducer Failure, Seal Housing SKP25647-1 Knife Edge Wear.

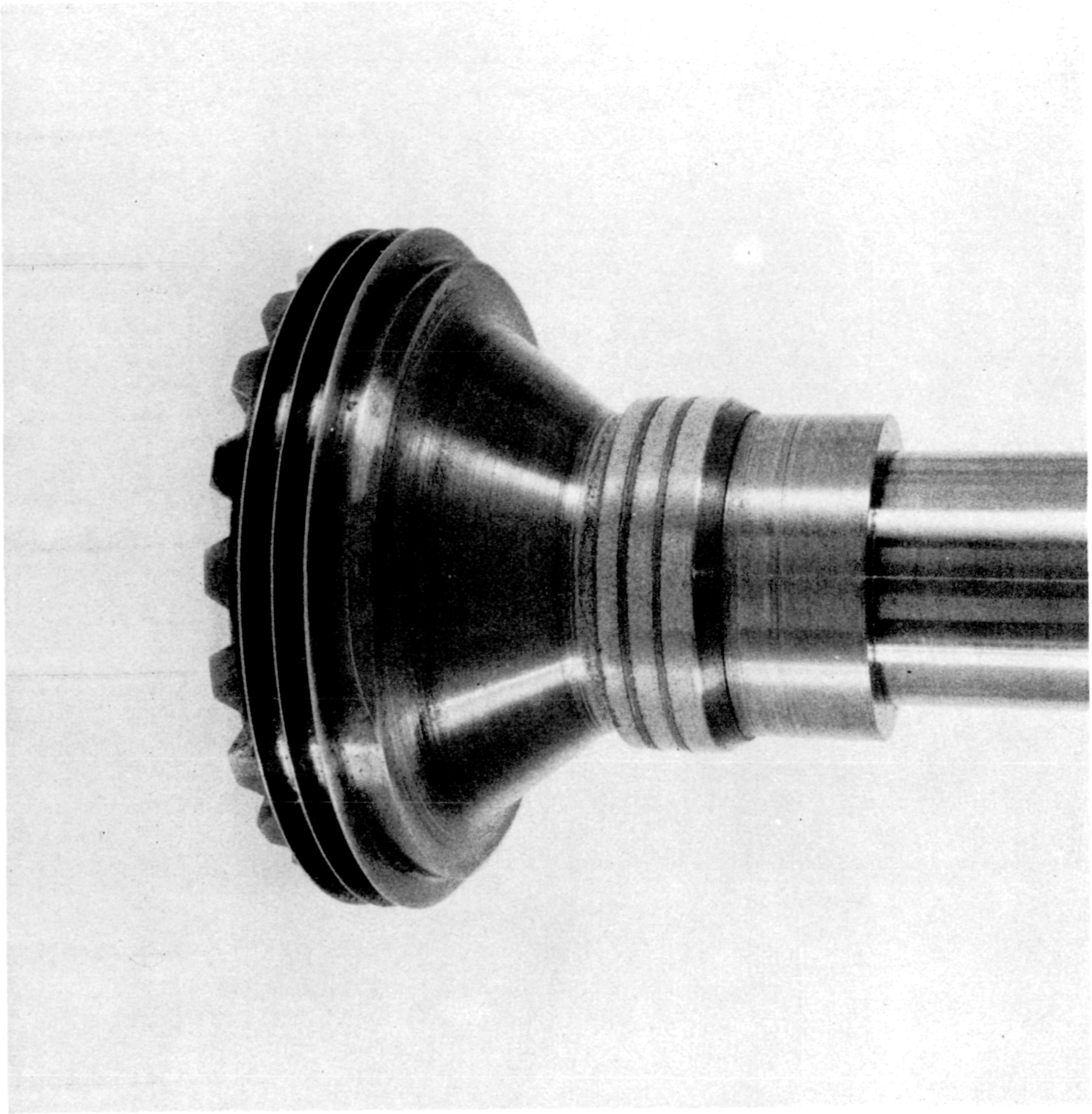


Figure 23. NASA 6:1 Compressor Test Rig, Inducer Failure, Turbine Shaft Seal Rub SKP25641-1.

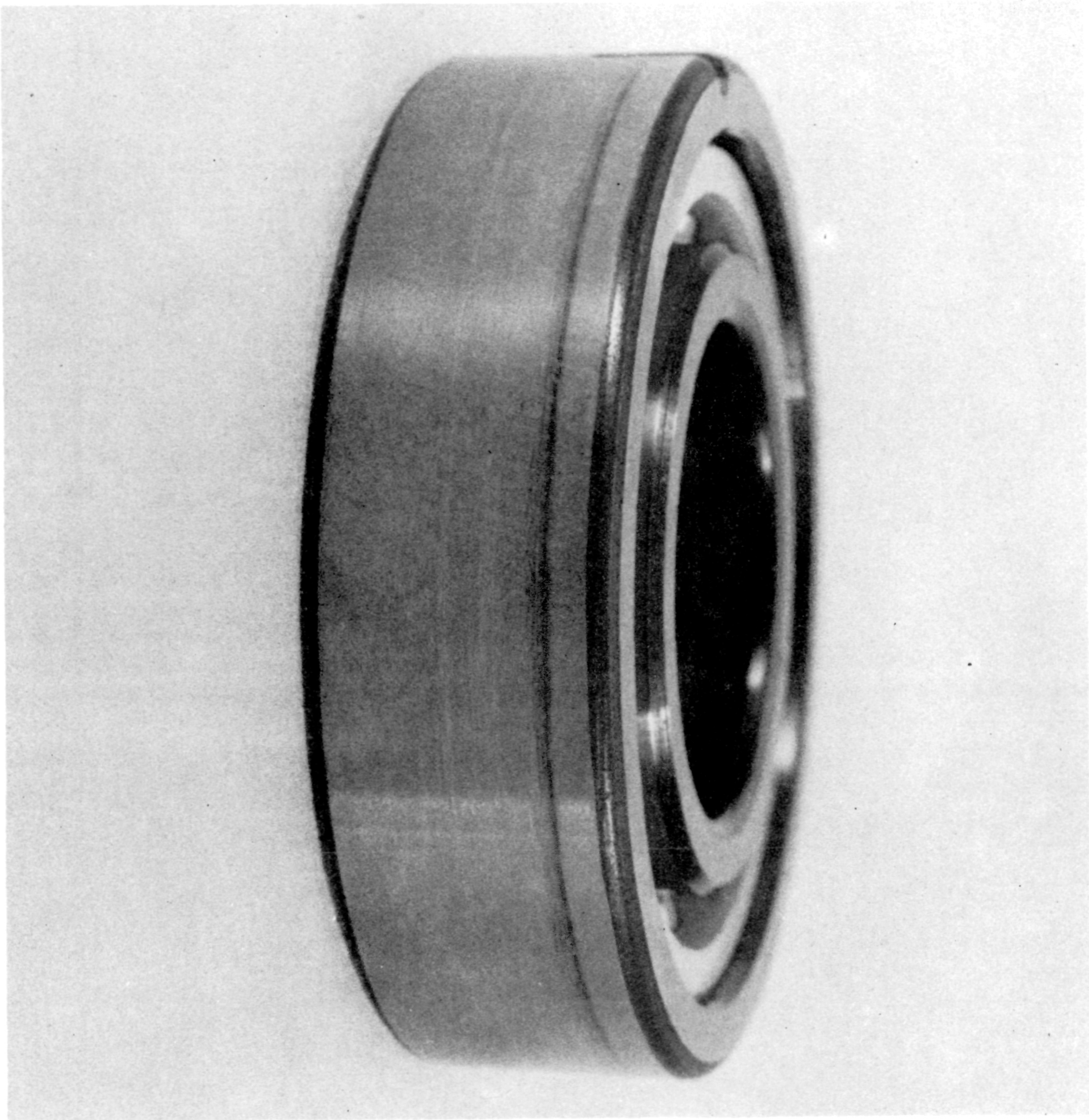


Figure 24. NASA 6:1 Compressor Test Rig, Inducer Failure, Bearing P/N 976693-1, Evidence of Outer Race Rotation.

In order to investigate the mechanical problems without further risk to the compressor blading a dummy mass was fabricated to replace the impeller and inducer. The mechanical properties of the hardware were presented in Table I of Appendix I. The test rig was assembled with the dummy mass and modified hardware. Modification consisted of removing 0.050 in. from the main shaft OD between bearing span, P/N SKP25641-1, removing 0.050 in. from the ID of bearing spacer P/N SKP25642-1, increasing the hydraulic mount clearance on the ID of the bearing sleeves from 0.003 to 0.005 in., and repinning the bearings to prevent rotation.

A NASA representative, Mr. Charles Pennington, witnessed Build 2B of the test rig. On March 27, 1972 the test rig successfully achieved a speed of 96,000 rpm. The maximum total shaft excursion noted during running was 0.0013-in. At 96,000 rpm this runout reduced to 0.001 in. total excursion.

As described in Appendix I, hand finish marks were implicated in the inducer blade failure. Electropolishing the inducer to remove surface hand finish marks was undertaken to allow testing to proceed with an impeller-inducer configuration prior to receipt of the new inducers. This would determine whether the rig was mechanically sound in the design configuration. After polishing and inspection, concurrance to proceed was obtained from the AiResearch Engineering Mechanics Group and the NASA-Lewis Project Manager.

Replacement inducers, SKP25657-1, were ordered because electropolishing to remove hand finish marks would alter blade profiles sufficiently to make them unacceptable.

Build No. 3

The test rig was assembled on May 3, 1972 to run the impeller to shroud clearance variation test and the 5-minute run at 120-percent design speed. The test was successfully completed on May 11.

Table VI shows the calibration data for the eight capacitance probes. Tables VII and VIII are the test data log sheets used during engine testing. The clearance probe test data from Table VIII and the calibration data from Table VI were used to plot the clearance versus speed curve shown in figure 25.

The gas bearing failed to operate freely at approximately 30,000 rpm. Examination revealed that the inlet air bearing strut, SKP25667-1, had contacted the inlet housing assembly, SKP25666-1.

The test rig successfully operated at 96,000 rpm (120 percent speed) for 5 minutes. Lissajous traces from the Bentley probes are shown as figures 26 and 27.

Y-317

TABLE VI.

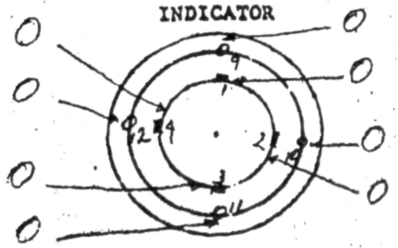
CLEARANCE PROBE CALIBRATION DATA
OUTPUT VOLTS AT INDICATED CLEARANCE
(VOLTAGE MUST BE ENTERED AS 4-DIGIT NUMBER)

		5	9	13	17	21	25	29	33	37	41	45	49	53	57	61	65	69	73	77
	PROBE	002	004	006	008	010	012	014	016	018	020	022	024	026	030	035	040	050	060	080
1ST STG RADIAL	1	2050	1550	1260	1060	0920	0820	0730	0680											
	2	1510	1230	1030	0900	0800	0720	0650	0600											
	3	2000	1480	1210	1030	0900	0800	0730	0670											
	4	1630	1300	1100	0950	0840	0750	0690	0650											
1ST STG INTER (TRUE)	5																			
	6																			
	7																			
	8																			
1ST STG AXIAL	9	2180	1870	1650	1450	1300	1170	1060	0980	0910	0840	0780	0730	0680	0600	0520	0450	0340	0270	0180
	10	2290	1950	1710	1500	1340	1210	1090	1000	0920	0850	0790	0740	0690	0610	0520	0460	0350	0270	0170
	11	1760	1550	1400	1250	1140	1030	0950	0870	0820	0760	0710	0670	0620	0550	0480	0420	0320	0250	0170
	12	2230	1890	1670	1460	1310	1180	1070	0980	0910	0840	0780	0730	0680	0600	0520	0450	0350	0270	0170
2ND STG RADIAL	13																			
	14																			
	15																			
	16																			
2ND STG INTER (TRUE)	17																			
	18																			
	19																			
	20																			
2ND STG AXIAL	21																			
	22																			
	23																			
	24																			

WHEEL: SKP 25657-1 / SKP 25658-1B S/N #2 DATE: 12-3-71 UNIT: NASA 6:1

SHROUD: SKP 25668-1 REC. BY: FULCHER CALIB. NO. Y 317

INDICATOR



FEELER

RUBBED AT IMPLR TIPS DIA

EQUIPMENT:

RUB OFF & 1-.008 2-.008 3-.008 4-.008

710 BALLANTINE S/N ______

KISTLER 56Y S/N ______

SINGLE CHANNEL CALIBRATOR
MOD. 722 S/N 1 WITH ______ PF
ADDED TO SIMULATE CELL

VOLTMETER ______ MOD ______
S/N ______ WITH ______ OHMS
IN PARALLEL ______ VOLT RANGE

NASA 6:1 COMPRESSOR RIG

DATE: 5/16/72
OPERATOR:
ASSISTANT:

1037   2:38:45   2:43:45

Speed	0		16,000		32,000		48,000		64,000	68,000	80,000	*	96,000	DOWN			
Oil Inlet Pressure PSIG			60		60		55	60	75	75	70	65	100				
Oil Inlet Temperature °F			107		126		164	177	142	180	220	248	118				
Oil Flow GPM			2.88		3.10		3.52	3.71	3.82	4.43	4.89	5.33	4.54				
Compressor Bearing Temperature °F																	
#1			102		130		175	190	165	200	235	260	160				
#2			103		130		175	190	175	202	240	270	160				
#3			103		130		175	190	165	200	235	262	150				
Turbine Bearing Temperature °F																	
#1			103		130		170	185	160	195	230	258	142				
#2			103		130		170	185	160	195	230	258	142				
#3			103		130		170	185	160	195	230	258	142				
Thrust																	
#1			75		70							✓					
#2			75		70												
#3			75		70												
Thrust Chamber Pressure PSIG			52		47							✓					
Vibration ~~(Diff)~~ g's																	
Vibration (Housing) g's			0		3.5g		15g	.15g	1.0g	.7g	.53g	10g					
Shaft Excursion																	
#1 Turbine																	
#2 Turbine																	
#1 Compressor																	
#2 Compressor																	
Turbine Inlet Temperature °F			60		49		210	210	280	350	387	415 ✓	515				
Turbine Inlet Pressure PSIG			3		10		25	25	54	54	96	88 ✓	160				
~~Turbine~~ Discharge Pressure PSIG												✓					
~~Diffuser~~ Force Lbs (TORQUE)			2		10	8	45	55	27			✓					
Gas Bearing Pressure (Top) PSIG			200		200		200	200	100	100	100	100	125				
Gas Bearing Pressure (Btm) PSIG			105		105		105	105	150	150	150	150	15				

* DO NOT EXCEED 5 MINUTES TOTAL OPERATION AT THIS SPEED. (✓ items Only)

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

NASA 6:1 COMPRESSOR TEST RIG

AiResearch Manufacturing Company of Arizona

TABLE VII.

SHEET 1

5/10/72

NASA 6:1 COMPRESSOR RIG

1007

Speed	0		16,000		32,000		48,000		64,000	68,000	80,000	*	96,000
~~Gas Bearing~~ Temperature (Btm) °F													
Compressor Discharge Pressure ~~PSIG~~	"Hg		PSIG 1		PSIG 2	PSIG 5	PSIG 6	PSIG 12	27.6	57.0	50.2	81.0 ✓	72
Compressor Discharge Temp °F			75		136	144	238	248	382	440	560	595	730
Clearance Probes VOLTS			✓		✓		✓		✓		✓		✓
#1 Radial			.095	.093	.102	.101	.104	.104	.101	.104	.101	.099	.101
#2			.086	.083	.079	.079	.084	.084	.088	.059	.081	.082	.093
#3			.109	.110	.105	.102	.103	.100	.107	.100	.106	.101	.—
#4			.093	.095	.107	.102	.099	.097	.096	.091	.105	.100	.—
#1 Axial 9			.033	.033	.034	.034	.036	.036	.039	.039	.042	.044	.051
#2 10			.036	.036	.038	.038	.040	.039	.042	.043	.045	.045	.—
#3 11			.032	.031	.033	.033	.035	.035	.038	.037	.040	.040	.043
#4 12			—	—	—	—	—	—	—	—	—	—	—
~~Compressor~~ Inlet Pressure													
#1 "H_2O													
#2													
#3													
BAROMETER IN. HgA	28.71		28.71	28.71					28.68	28.68	28.68	28.68	28.68
BELL MOUTH STATIC PRESS. IN. H_2O			.4	.2	2.2	1.3	5.5	4.6	11.9	11.0	22.6	20.0	32
COMPRESSOR INLET TEMP °F			75	75	74	75	96	96	104	128	144	158	115

* DO NOT EXCEED 5 MINUTES TOTAL OPERATION AT THIS SPEED (ONLY ✓ DATA)

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

NASA 6:1 COMPRESSOR TEST RIG

TABLE VIII.

AiResearch Manufacturing Company of Arizona

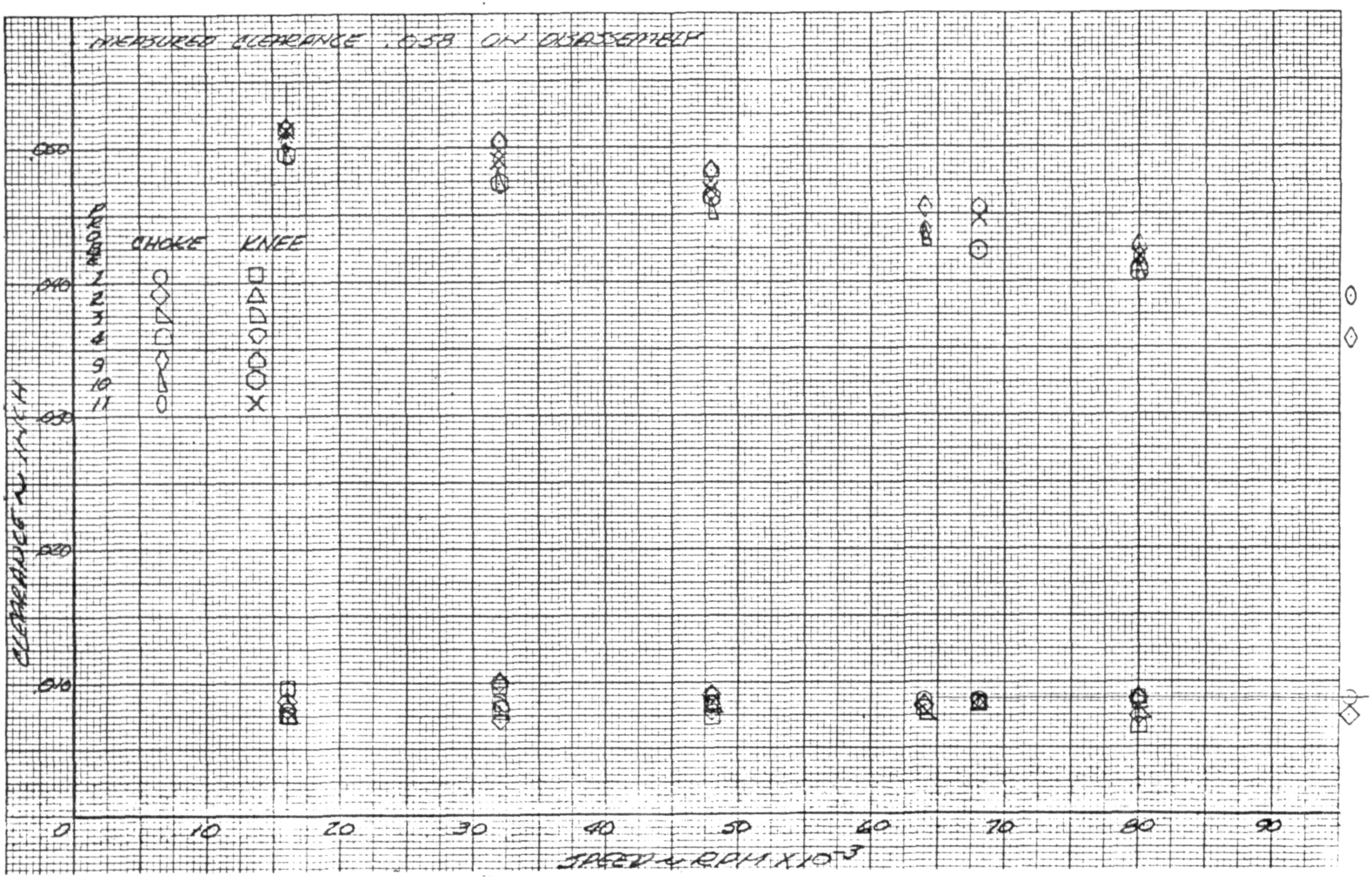
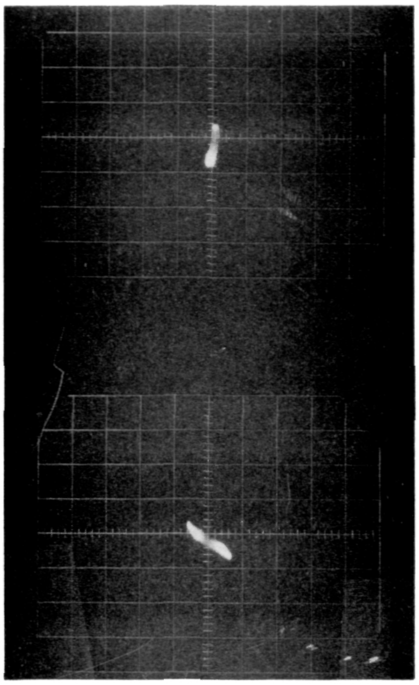


Figure 25. Capacitance Probe Clearance.

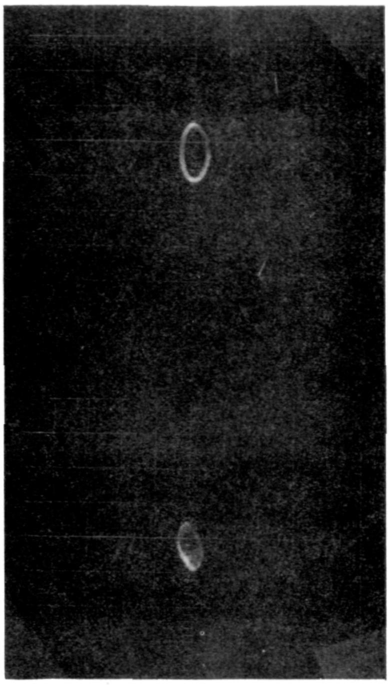
NASA 6:1 COMPRESSOR RIG
BUILD 3 LISSAJOUS TRACES

COMPRESSOR
END

TURBINE
END



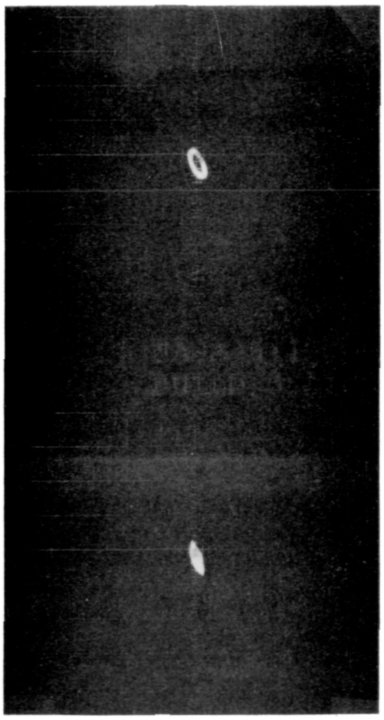
16,300 RPM



32,090 RPM

COMPRESSOR
END

TURBINE
END



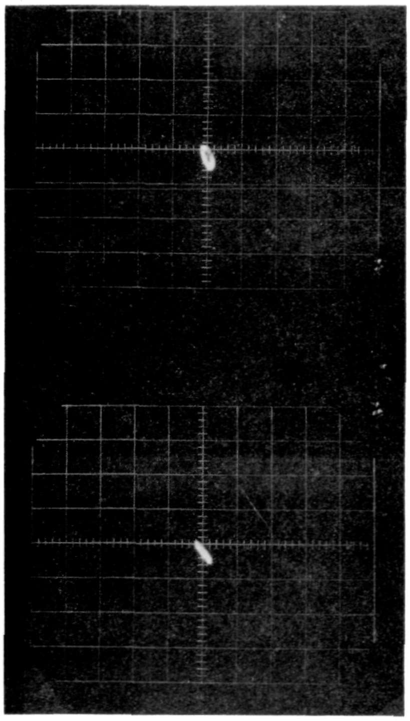


Figure 26.

48,400 RPM

64,400 RPM

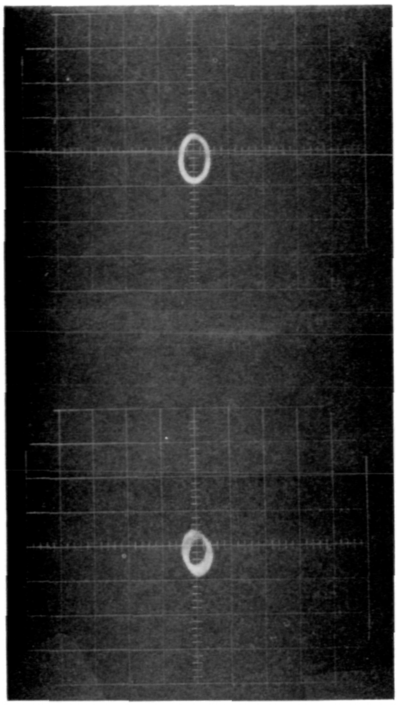
LISSAJOUS SCALES: 1.0 MILS/DI
.2 VOLTS/DI
TEST DATE: 11 MAY 1972

MP-37107

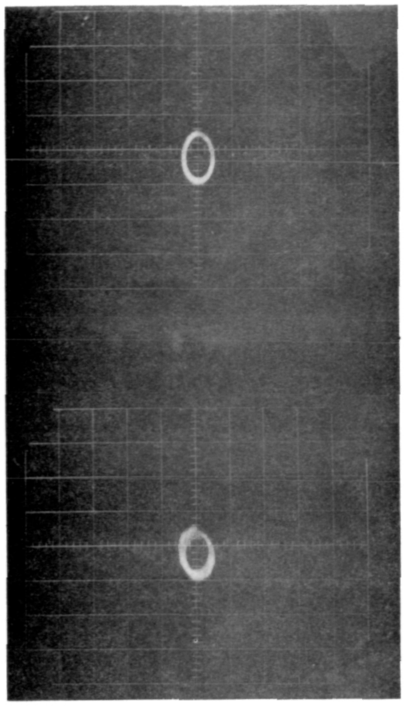
NASA 6:1 COMPRESSOR RIG
BUILD 3 LISSAJOUS TRACES

COMPRESSOR END

TURBINE END



80,450 RPM

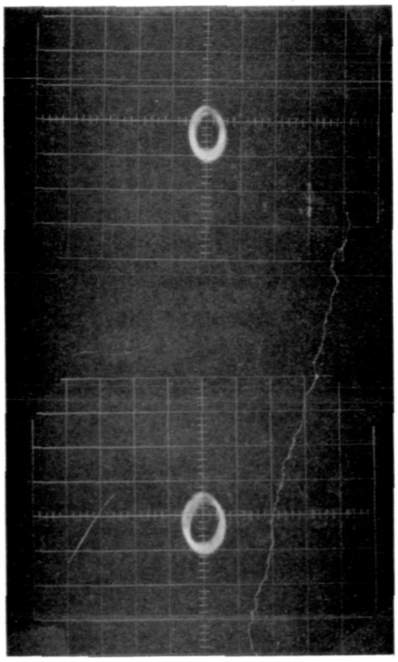


88,600 RPM

5-MINUTE OVERSPEED TEST

COMPRESSOR END

TURBINE END



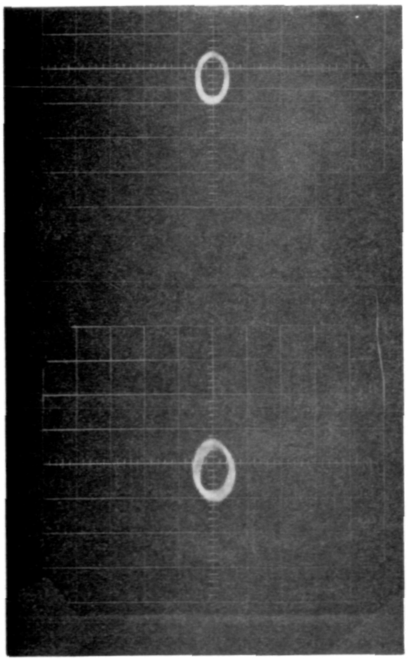


Figure 27.

96,000 RPM
START OF TEST

96,000 RPM
END OF TEST

LISSAJOUS SCALES: 1.0 MILS/DIV
0.2 VOLTS/DIV

TEST DATE: 11 MAY 1972

Disassembly of the test rig revealed that the front side of the turbine wheel had rubbed the nozzle housing. Figures 28 through 42 show test rig parts. The inducer and impeller were zygloed and the turbine wheel was magnafluxed. No indications of distress were observed. All other hardware appeared in excellent condition except the strain gage had a stray ground. The part was sent to instrumentation and repaired.

Stress review of the gas bearing area resulted in redesign of the inlet air bearing housing, SKP25667-1. The clearance and the supply orifices were increased in size resulting in a revised estimated airflow of 0.10 lb/sec.

Build 4 and 4A

Build 4 of the test rig was assembled to the dimensions shown in Table IX, installed in the test cell, and operated on June 27, 1972, to check the instrumentation and gas bearings. At 32,000 rpm, oil began spraying from the rig. Examination of the rig indicated that the seal on the compressor side had failed. Also, the gas bearing was not electrically isolated from the remainder of the rig as indicated by instrumentation prior to the run. The rig was removed from the test cell and was returned to development assembly for repair. The compressor-side carbon seal was replaced. Test data taken is presented in Table X. Lissajous traces from the Bentley probes are shown in figure 43.

On June 30, 1972, the rig was again installed and operated in the test cell. The test was terminated because of excessive oil leakage and instrumentation showed gas bearing continuity.

Disassembly of the rig revealed that the turbine-side secondary Teflon ring seal had broken into two pieces. The cause was attributed to excessively high pressure applied behind the turbine wheel extruding the Teflon ring past its locking device. To prevent recurrance of this failure, a retaining ring was fabricated to provide a larger face contact area for the Teflon seal.

The compressor discharge knife-edge seal on the turbine side measured 0.0045 in. out of flat which allowed the measured clearance to close down to 0.001 in. The decrease in clearance was likely the cause of the inability to electrically isolate the gas bearing after assembly. The seal was reworked to bring the flatness to within the required 0.001 in. maximum.

The SKP25657-1 inducers were received from the vendor prior to final finishing. The inducers were reviewed by AiResearch, were found to be acceptable, and were returned to the vendor for final finishing.

Figure 28. Shroud Assembly, NASA 6:1 Compressor Test Rig, 120-Percent Speed Test.

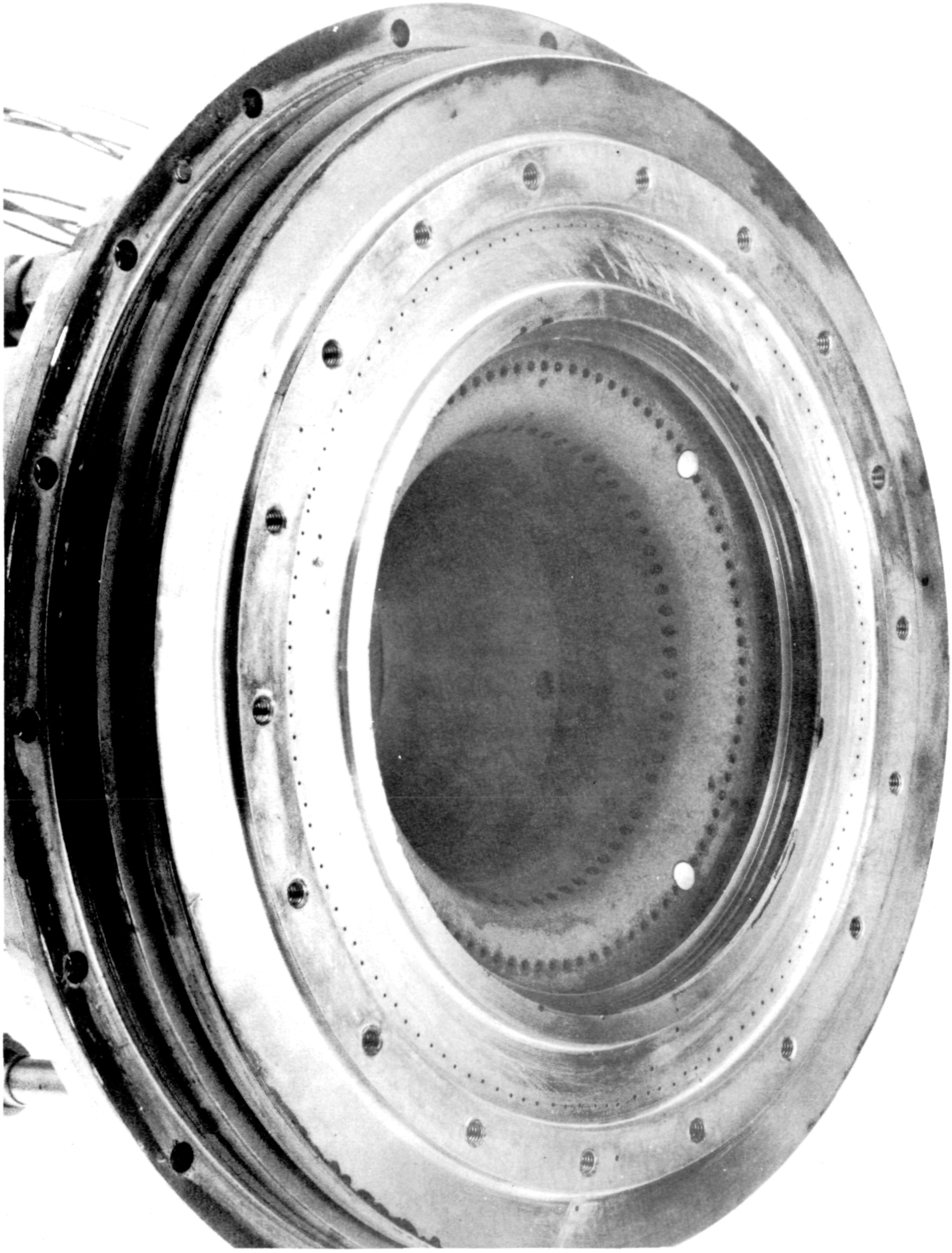


Figure 29. Shroud Assembly, NASA 6:1 Compressor Test Rig, 120-Percent Speed Test.

Figure 30. Impeller SKP25658-1,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

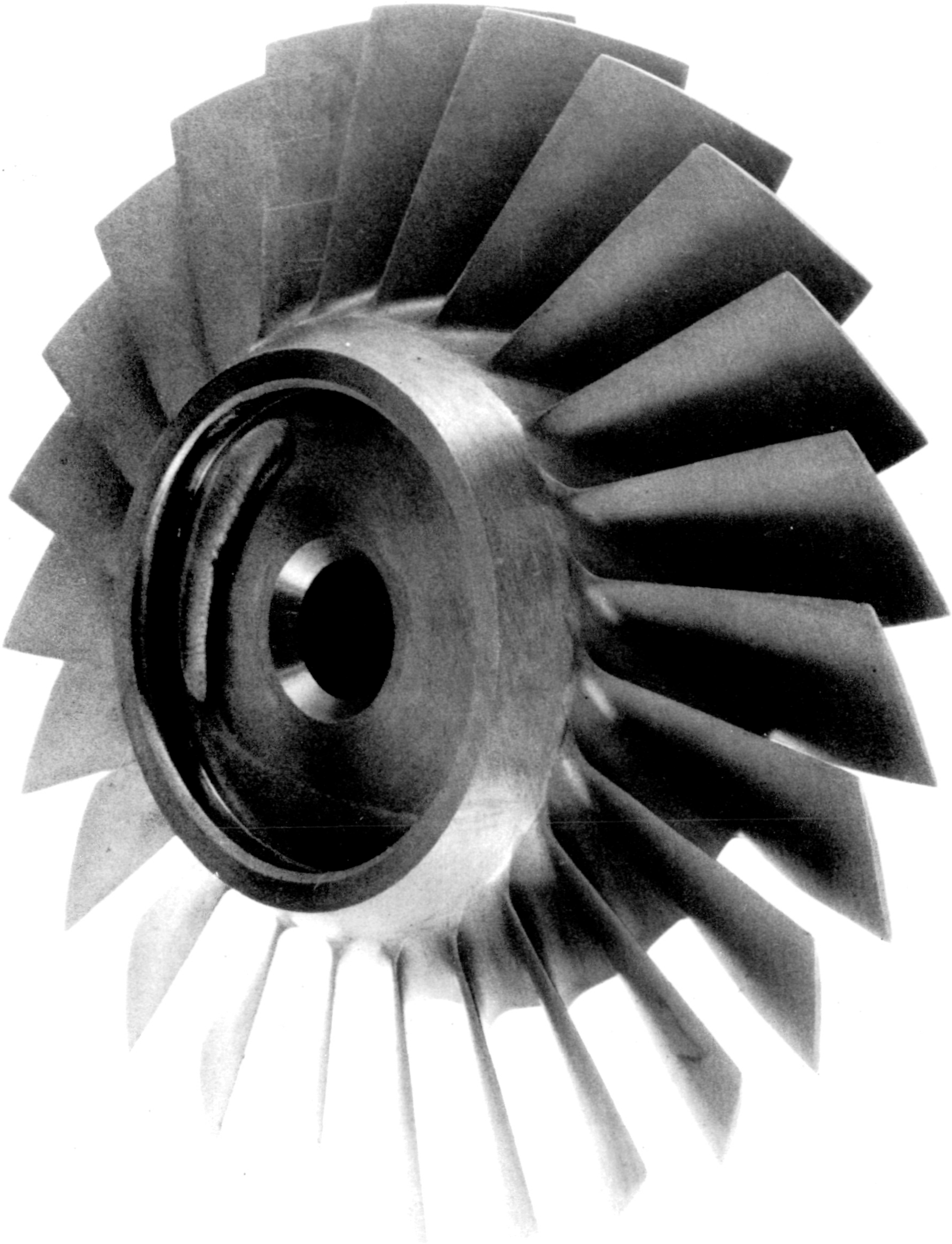


Figure 31. Inducer SKP25657-1, NASA 6:1 Compressor Test Rig, 120-Percent Speed Test.

Figure 32. Retainer SKP25643-1,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

MP-34169

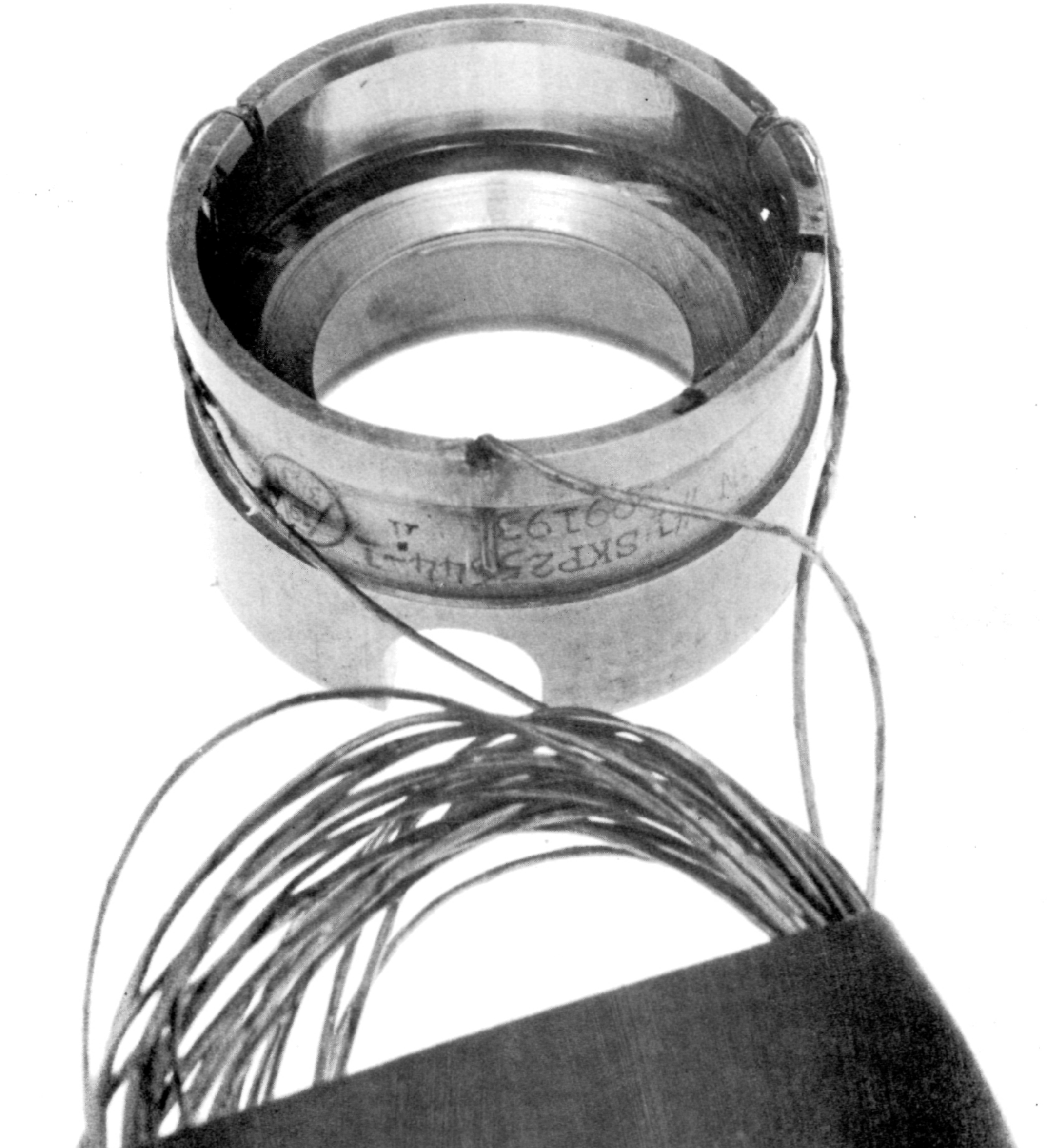


Figure 33. Retainer SKP25644-1, NASA 6:1 Compressor Test Rig, 120-Percent Speed Test.

Figure 34. Seal Plate 25669-1,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

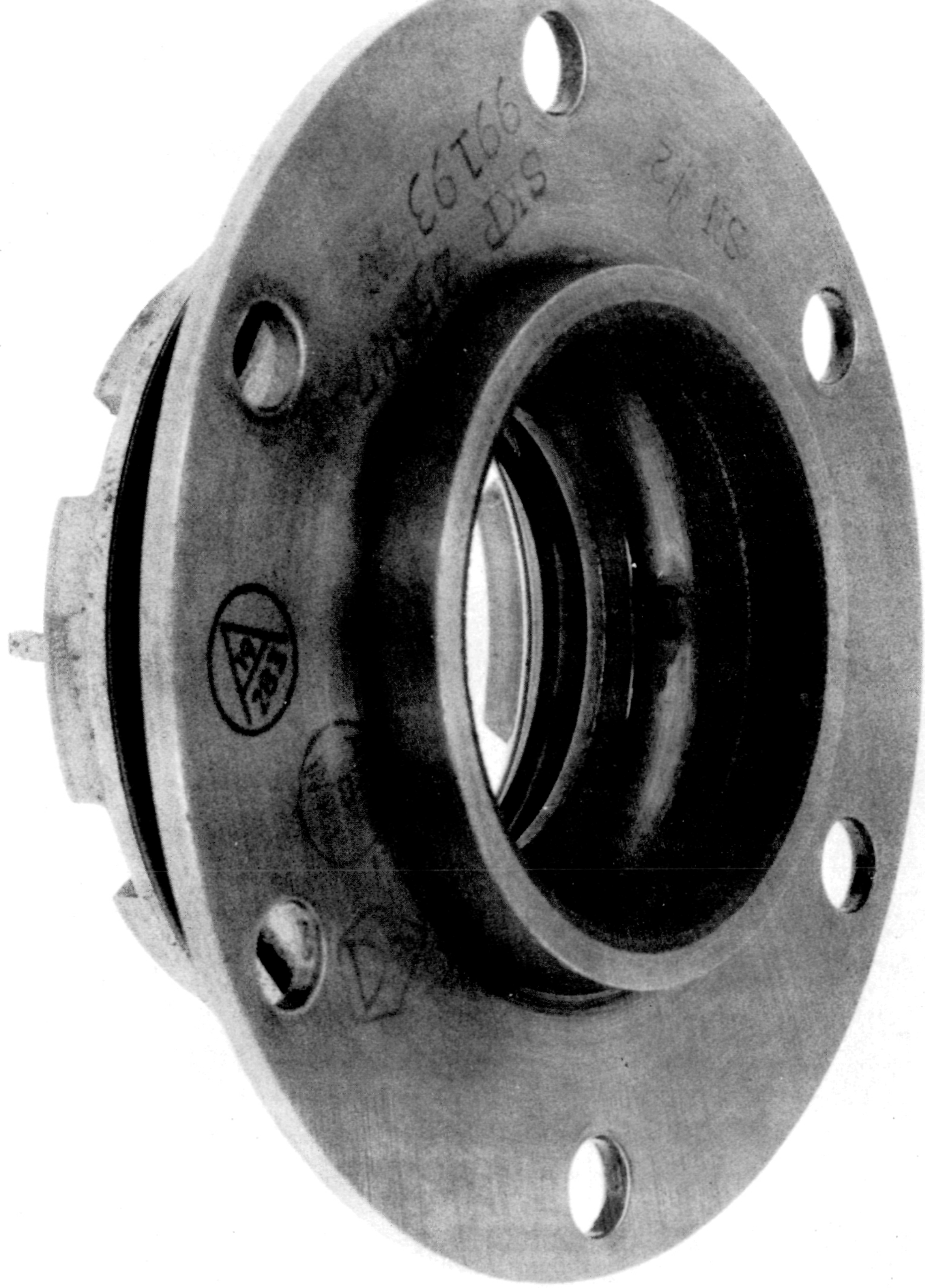


Figure 35. Seal Retainer SKP25647-1,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

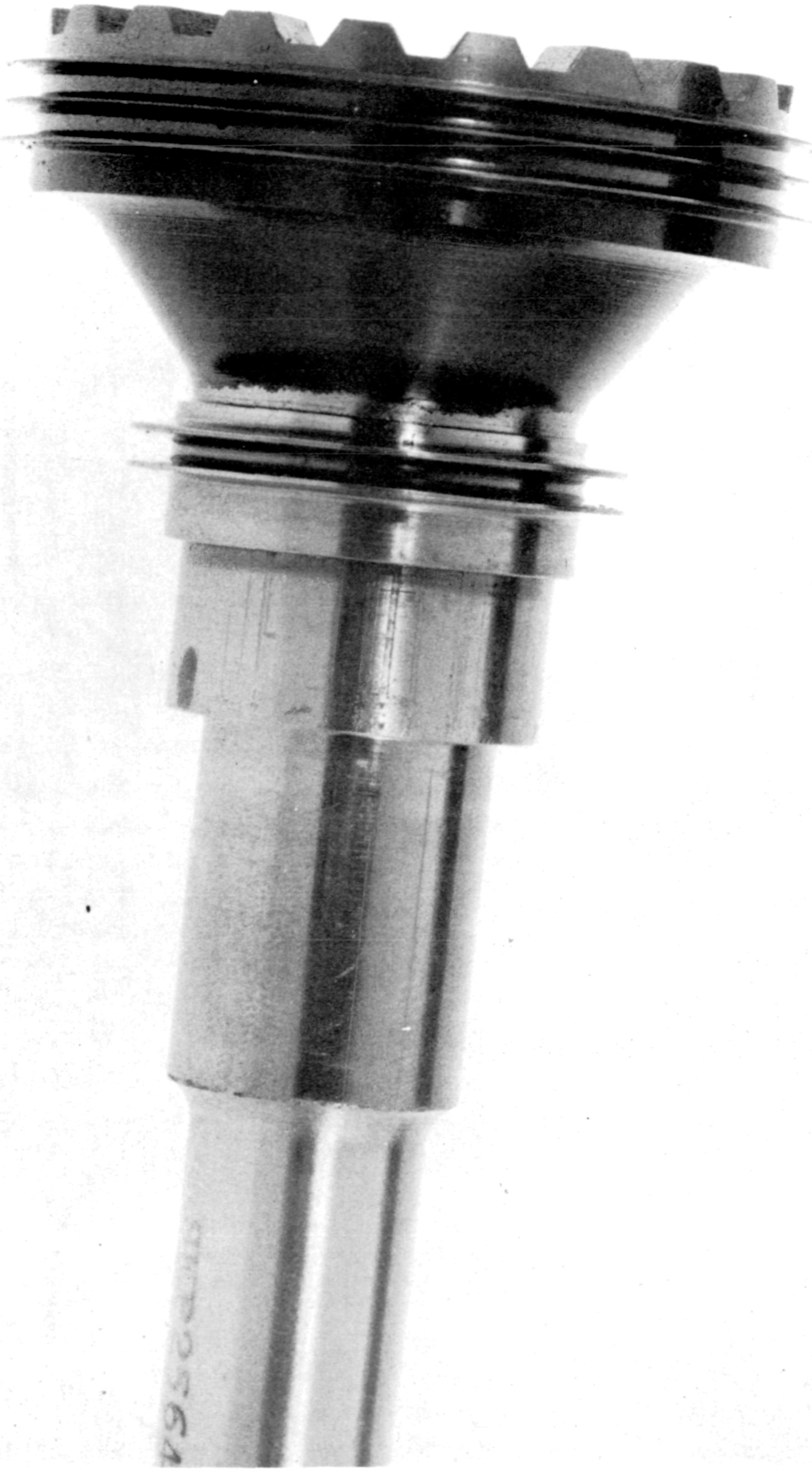


Figure 36. Shaft SKP25641-1,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

Figure 37. Bearing 976693-1,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

Figure 38. Bearing 976693-1, NASA 6:1 Compressor Test Rig, 120-Percent Speed Test.

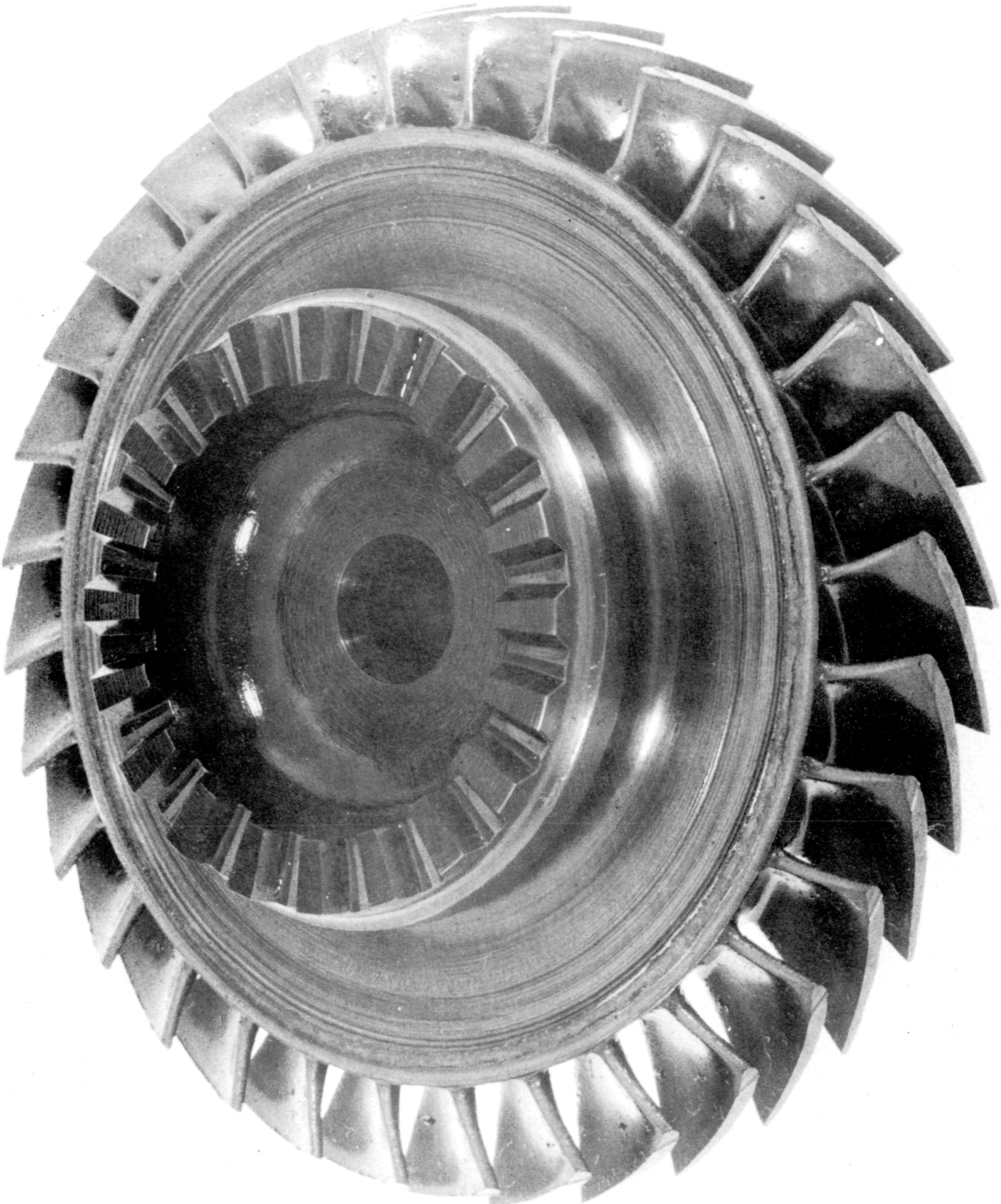


Figure 39. NASA Turbine Wheel,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

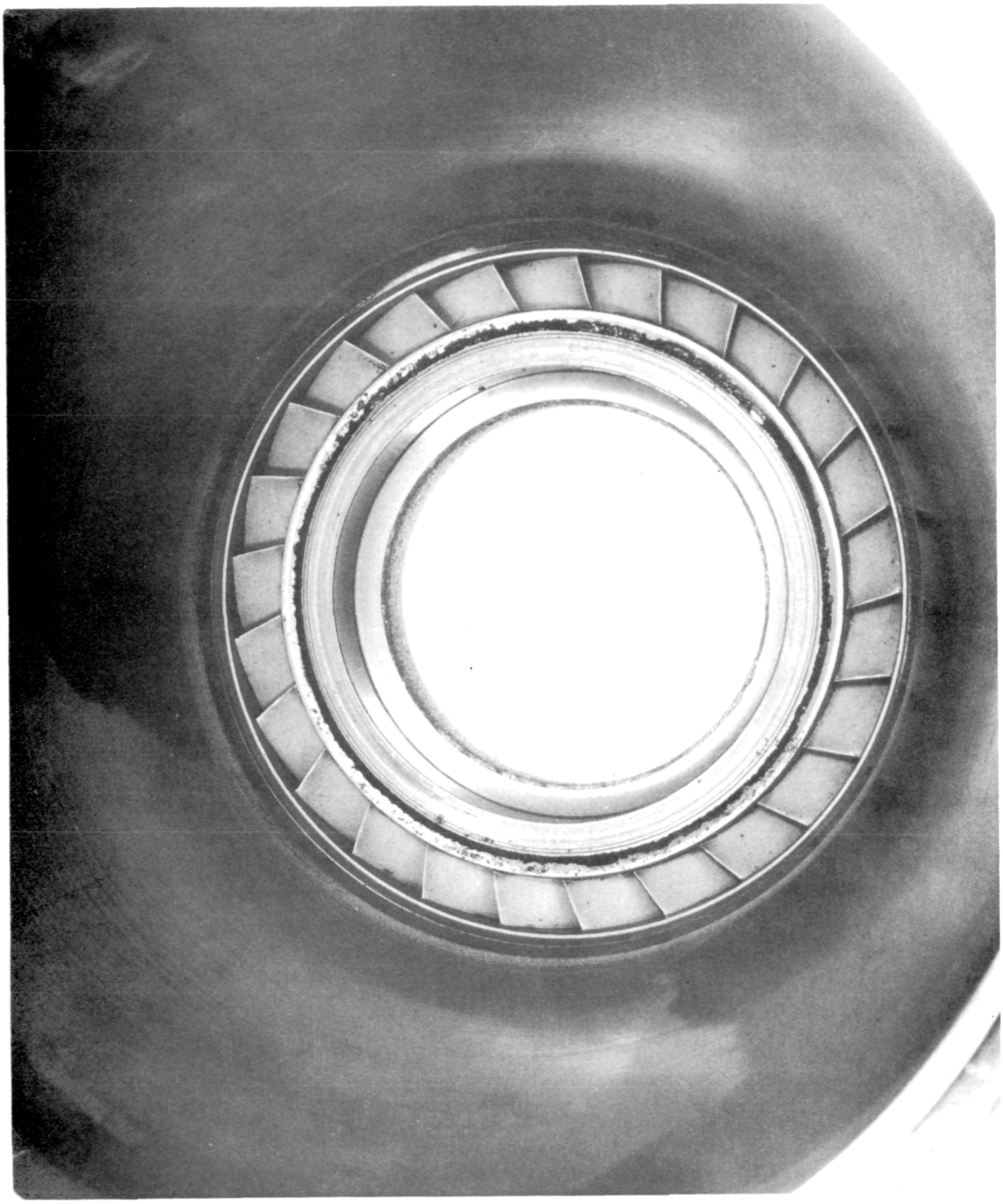


Figure 40. NASA Nozzle Assembly,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

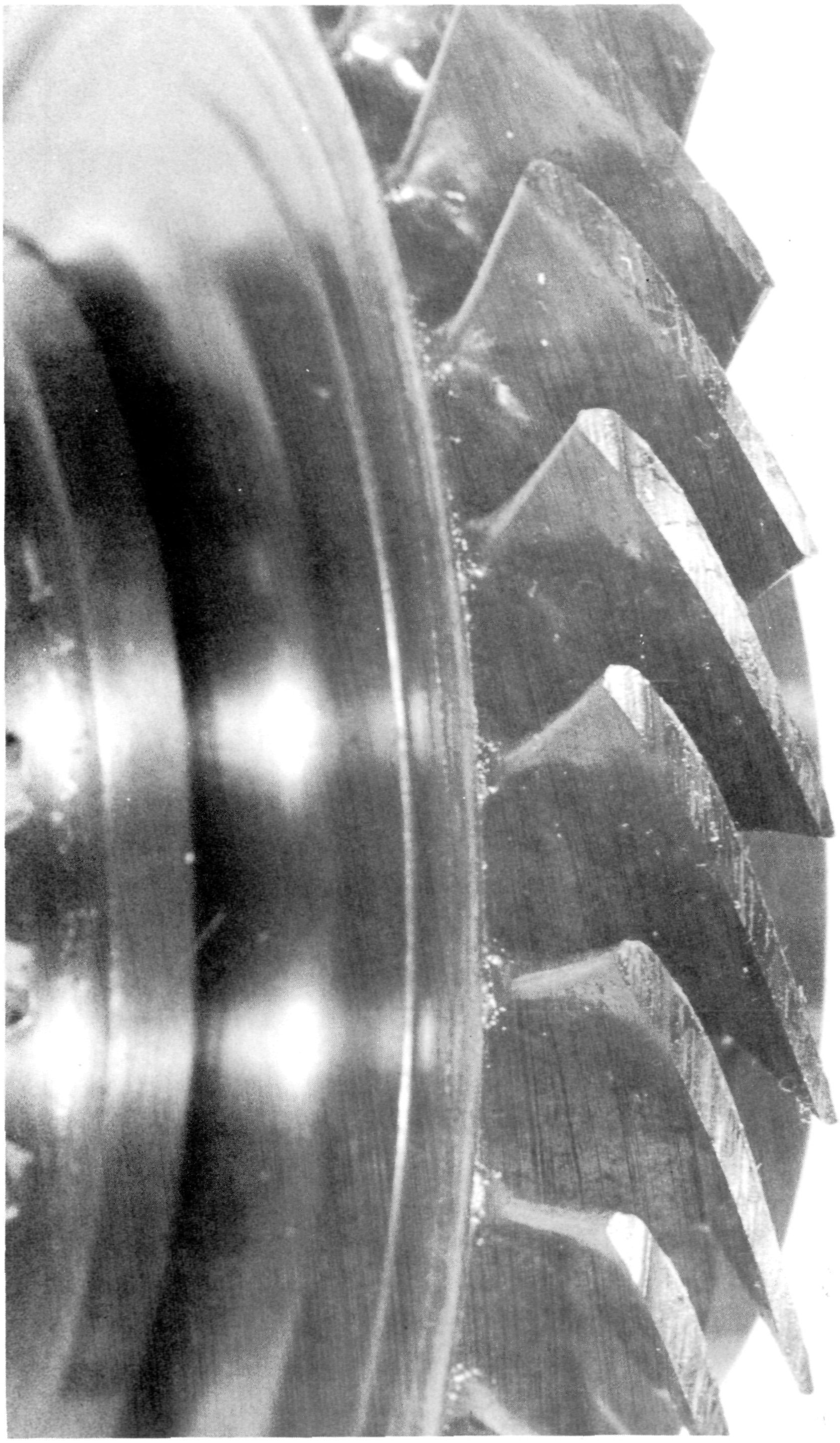


Figure 41. NASA Turbine Wheel, NASA 6:1 Compressor Test Rig, 120-Percent Speed Test.

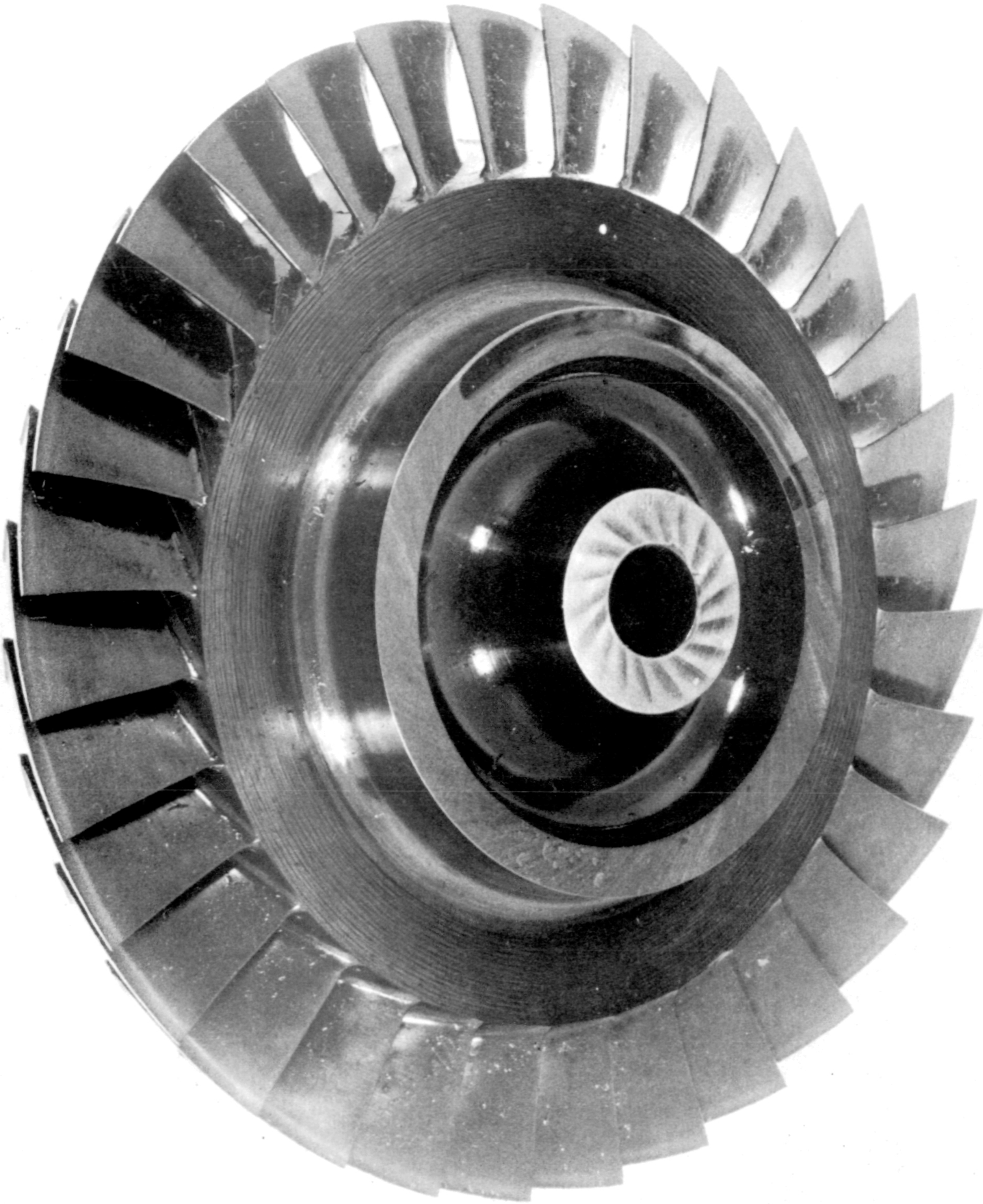


Figure 42. NASA Turbine Wheel,NASA 6:1 Compressor Test Rig,120-Percent Speed Test.

TABLE IX.

BUILD 4 DATA SHEET
NASA 6:1 COMPRESSOR

A. Runout

1. Turbine

a.	O.D.	0.0012
b.	Front Face	0.0007
c.	Knife Edge Seal	0.0007

2. Compressor

a.	O.D.	0.0008
b.	Back Face	0.0008
c.	Knife Edge Seal	0.001

B. Balance	Max Allowed	Actual
1. Turbine	0.017 Oz-In.	0.010
2. Compressor	0.023 Oz.-In.	0.010

C. Clearances	B/P	Actual
1. Turbine Bearing Housing to Seal Retainer	0.023-0.027	0.025
2. Compressor Bearing Housing to Seal Retainer	0.001-0.003	0.003
3. Diffuser Knives to Discharge, Turbine End	0.003-0.005	0.003
4. Diffuser Knives to Discharge, Compressor End	0.003-0.005	0.006
5. Turbine Wheel Clearance	0.023-0.027	0.058
6. Compressor Face Clearance (with floating diffuser)	0.021-0.023	0.021

FORM P-702

NASA 6:1 COMPRESSOR RIG

DATE: 6/27/72
OPERATOR: STEWART
ASSISTANT: FRICKE

Speed	0		16,000		32,000		48,000		64,000		80,000	*	96,000			
Oil Inlet Pressure PSIG			65		75											
Oil Inlet Temperature °F			89		85											
Oil Flow ~~GPM~~ CPS			700		780											
Compressor Bearing Temperature °F																
#1			95		100											
#2			95		100											
#3			95		100											
Turbine Bearing Temperature °F																
#1			95		100											
#2			95		100											
#3			95		100											
Thrust																
#1												✓				
#2																
#3																
Thrust Chamber Pressure PSIG												✓				
Vibration (Diff) g's			0		.01											
Vibration (Housing) g's																
Shaft Excursion																
#1 Turbine																
#2 Turbine																
#1 Compressor																
#2 Compressor																
Turbine Inlet Temperature °F			68		55							✓				
Turbine Inlet Pressure PSIG			10		12							✓				
Turbine Discharge Pressure PSIG												✓				
Diffuser Force Lbs			3		5							✓				
JOURNAL Gas Bearing Pressure (Top) PSIG			24		20											
Gas Bearing Pressure (Btm) PSIG																

* DO NOT EXCEED 5 MINUTES TOTAL OPERATION AT THIS SPEED. (✓ items Only)

ALTITUDE EQUIPMENT DIVISION
CALCULATED
RECORDED
DRAWN
CHECKED
APPROVED

NASA 6:1 TEST RIG
BUILD NO. 4

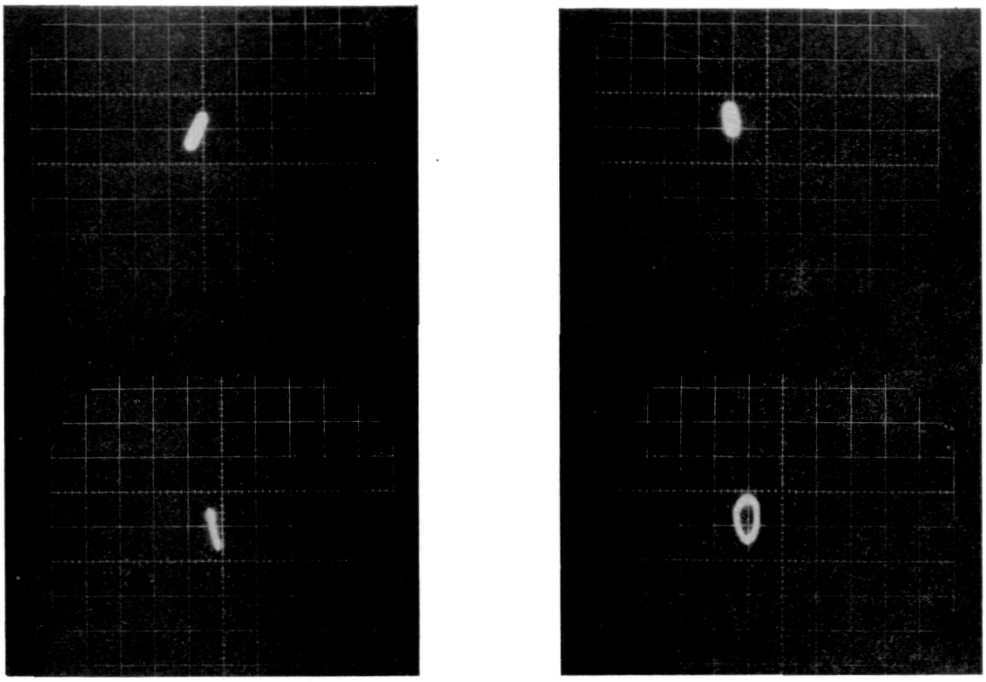
AiResearch Manufacturing Company of Arizona

TABLE X.

NASA 6:1 COMPRESSOR RIG
BUILD 4 LISSAJOUS TRACES

COMPRESSOR
END

TURBINE
END



16,390 RPM

32,290 RPM

SCALES: 1.0 MIL/DIV
0.2 VOLT/DIV

TEST DATE: 27 JUNE 1972

Figure 43.

On August 11th, Build 4A of the compressor rig assembly was completed and ready for testing. The test setup was designed for resistance measurement readings to be available across the floating diffuser/shroud assembly. Also, two actuators were mounted on the front bearing carrier such that circumferential torque loads could be applied to the diffuser in either direction. Between the electrical continuity check and the torque hysteresis reading on the Brown Torquemeter, the degree of accuracy available from future torque or horsepower performance data could be determined. Build dimensions for this assembly are given in Table XI.

The rig was operated with the following torque results before and after diffuser actuation.

Speed, rpm	Torque (before/after), in.-lb
zero	zero
40 K	124/124
50 K	215/215
60 K	335/335
70 K	480/480
75 K	Diffuser sticks
80 K	Diffuser sticks

Heated air to 375°F was applied to the gas thrust bearing to determine if the diffuser had thermally distorted to cause an electrical ground to the case. Neither heated air nor speed variations were sufficient to create a floating diffuser satisfactory for performance testing.

At 80,000 rpm, the turbine seal began leaking oil. Since the secondary seal or Teflon tech rings had previously created seal problems, they were suspected again to be sealing inadequately. The turbine thrust piston cavity pressure was raised to 60 psig. This was determined adequate to provide several pounds of pressure to the Teflon secondary seal to stop oil leakage at the expense of raising the ball thrust bearing load toward the turbine. Test data is recorded in Table XII and lissajous traces from the Bentley proximity probes are shown in figure 44.

TABLE XI.

BUILD 4A DATA SHEET
NASA 6:1 COMPRESSOR

A. Runout

1. Turbine

a.	O.D.	0.0012
b.	Front Face	0.0007
c.	Knife Edge Seal	0.0007

2. Compressor

a.	O.D.	0.0008
b.	Back Face	0.0008
c.	Knife Edge Seal	0.001

B. Balance	Max Allowed	Actual
1. Turbine	0.017 Oz-In.	0.010
2. Compressor	0.023 Oz.-In.	0.010

C. Clearances	B/P	Actual
1. Turbine Bearing Housing to Seal Retainer	0.023-0.027	0.025
2. Compressor Bearing Housing to Seal Retainer	0.001-0.003	0.003
3. Diffuser Knives to Discharge, Turbine End	0.003-0.005	0.005
4. Diffuser Knives to Discharge, Compressor End	0.003-0.005	0.008
5. Turbine Wheel Clearance	0.023-0.027	0.058
6. Compressor Face Clearance	0.021-0.023	0.024

FORM P-799

NASA 6:1 COMPRESSOR RIG

GAS BR.

DATE: 8-8-72 A.M.
OPERATOR: E. Stuart
ASSISTANT: B. Fricke

Speed	0		16,000		32,000	40,000	~~43,000~~ 50,000	60,000	64,000	70,000	80,000	*	96,000			
Oil Inlet Pressure PSIG						65	65	65		60						
Oil Inlet Temperature °F						95	112	128		160						
Oil Flow GPM						4.5	4.1	5.5		9.1						
Compressor Bearing Temperature °F						40										
#1						110	120	150		240						
#2						110	120	150		240						
#3							120	150		240						
Turbine Bearing Temperature °F																
#1						115	140	150		275						
#2						115	140	150		280						
#3						115	140	150		280						
Thrust																
#1												✓				
#2																
#3																
Thrust Chamber Pressure PSIG						10	12	15		5		✓				
Vibration (Diff) g's																
Vibration (Housing) g's																
Shaft Excursion																
#1 Turbine																
#2 Turbine																
#1 Compressor																
#2 Compressor																
Turbine Inlet Temperature °F						200	355	325	320			✓				
Turbine Inlet Pressure PSIG						16	26	40	68			✓				
Turbine Discharge Pressure PSIG						—	—					✓				
Diffuser Force Lbs						255/275	375/400	580/600	325/335			✓				
Gas Bearing Pressure (Top) PSIG																
Gas Bearing Pressure (Btm) PSIG																

*. DO NOT EXCEED 5 MINUTES TOTAL OPERATION AT THIS SPEED. (✓ items Only).

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

NASA 6:1 TEST RIG
BUILD NO. 4A.

AiResearch Manufacturing Company of Arizona

TABLE XII.

89

NASA 6:1 COMPRESSOR RIG

	Speed	0		16,000		32,000	40,000	~~40,000~~ 50,000	60,000	64,000	70,000	80,000	*	96,000			
	Gas Bearing Temperature (Btm) °F								350°								
"Hg	Compressor Discharge Pressure PSIG						12.4	200	30.0		42.4		✓				
	Compressor Discharge Temp °F						175	250	332		440		✓				
27 1	Clearance Probes						85	115	126		161						
28 2	#1 Radial						80	103	119		140		✓				
29 3	#2						82	110	133		140		✓				
30 4	#3						73	94	104		114		✓				
31 5	#4						74	92	104		129		✓				
32 6	#1 Axial						70	86	94		114		✓				
33 7	#2						8	—	—		—		✓				
34 8	#3						14[illegible]	216	273		380		✓				
35 9	#4						14[illegible]	210	274		361		✓				
36 10							16[illegible]	243	326		426						
7	Compressor Inlet Pressure						16[illegible]	246	328		434						
8	#1 "H_2O						16[illegible]	243	328		448						
9	#2						8[illegible]	116	134		287						
	#3																

* DO NOT EXCEED 5 MINUTES TOTAL OPERATION AT THIS SPEED (ONLY ✓ DATA)

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

NASA 6:1 TEST RIG
BUILD NO. 4A

AiResearch Manufacturing Company of Arizona

TABLE XII. (Contd)

															DATE:	8-10-72
NASA 6:1 COMPRESSOR RIG															OPERATOR:	E. [illegible]
															ASSISTANT:	R. [illegible]
Speed	0				40,000		50,000		60,000		70,000		80,000			
Oil Inlet Pressure PSIG					65		65		65		65		65			
Oil Inlet Temperature °F					91		95		112		134		185			
Oil Flow GPM					4.2		4.6		4.5		5.5		11.5			
Compressor Bearing Temperature °F																
#1					110		140		140		145		220			
#2					110		140		140		145		215			
#3					110		140		140		145		215			
Turbine Bearing Temperature °F																
#1					—		—		—		—		—			
#2					120		145		200		210		230			
#3					120		120		140		160		210			
Thrust																
#1							70		68		52		105			
#2																
#3																
Thrust Chamber Pressure PSIG							42		63		110		15			
Vibration (Diff) g's																
Vibration (Housing) g's																
Shaft Excursion																
#1 Turbine																
#2 Turbine																
#1 Compressor																
#2 Compressor																
Turbine Inlet Temperature °F					150		290		370		460		390			
Turbine Inlet Pressure PSIG					20		30		45		75		118			
Turbine Discharge Pressure PSIG					—		—		—		—		—			
Diffuser Force Lbs					160/160	[illegible]	310/310		505/510		790/830					
Gas Bearing Pressure (Top) PSIG											760/960 (Stuck)					
Gas Bearing Pressure (Btm) PSIG																

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

NASA 6:1 TEST RIG
BUILD NO. 4A

AiResearch Manufacturing Company of Arizona

TABLE XII. (Contd)

8-10-72

NASA 6:1 COMPRESSOR RIG

Speed	0				40,000		50,000		60,000		70,000	INCHES	80,000	INCHES		
Gas Bearing Temperature (Btm) °F																
Compressor Discharge Pressure PSIG					11.8		19.8		31.1		48.0		65.5			
Compressor Discharge Temp °F					167		220		298		280		442			
Clearance Probes																
#1 Radial											.115	<.005	.131-.137	≈.003		
#2								SEE CALIBRATION			.083	<.010	.084-.087	≈.003-.005		
#3											.096	.009	.099-.100	≈.0085		
#4								Y-317			.126-.130	≈.004	.117-.123	.005±.0005		
10 #1 Axial											.070	.026	.076	.023		
11 #2								NOTE: AXIAL CLEAR.			.065	.025	.072	.022		
~~12~~ #3								CHGD WITH TIME –								
#4								MAY BE .005" LESS								
								IF ROTOR IS BROUGHT								
Compressor Inlet Pressure								UP TO SPEED QUICKLY								
#1 "H_2O																
#2																
#3																
T/C → 27					80		84		86		88		70			
28					80		86		90		84		68			
29					80		86		86		88		72			
30					78		82		83		73		54			
31					76		80		78		64		46			
32					78		82		84		76		57			
33					—		—		—		—		—			
34					148		186		244		312		349			
35					132		162		197		246		371			
36					166		220		274		355		416			
37					166		222		274		355		424			
38					166		220		276		358		410			
39					90		103		130		220		280			

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

NASA 6:1 TEST RIG
BUILD NO. 4A

AiResearch Manufacturing Company of Arizona

TABLE XII. (Contd)

NASA 6:1 COMPRESSOR RIG

DATE: 8-11-72
OPERATOR: E. [illegible]
ASSISTANT: B. [illegible]

Speed	0	40,000	50,000	60,000	70,000	80,000				
Oil Inlet Pressure PSIG		70	70	65	65	65				
Oil Inlet Temperature °F		115	120	122	140	175				
Oil Flow GPM										
Compressor Bearing Temperature °F										
#1		115	120	150	140	200				
#2		115	130	150	140	200				
#3		115	130	150	140	195				
Turbine Bearing Temperature °F										
#1		—	—	—	—	—				
#2		110	160	150	275	260				
#3		75	110	205	159	205				
Thrust										
#1		120	70	175	105	90				
#2										
#3										
Thrust Chamber Pressure PSIG		2	4	6.5	9.5	13				
Vibration (Diff) g's										
Vibration (Housing) g's										
Shaft Excursion										
#1 Turbine										
#2 Turbine										
#1 Compressor										
#2 Compressor										
Turbine Inlet Temperature °F		190	275	340	415	425				
Turbine Inlet Pressure PSIG		15	30	45	70	105				
Turbine Discharge Pressure PSIG										
Diffuser Force Lbs		0/117	180/[illegible]	[illegible]/32[illegible]	3[illegible]/70	Locked up				
Gas Bearing Pressure (Top) PSIG					[illegible]					
Gas Bearing Pressure (Btm) PSIG										

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

NASA 6:1 TEST RIG
BUILD NO. 4A

AiResearch Manufacturing Company of Arizona

TABLE XII. (Contd)

NASA 6:1 COMPRESSOR RIG

8-11-72

Speed	0		40,000		50,000		60,000		70,000	INCHES	80,000	INCHES
Gas Bearing Temperature (Btm) °F			75				68		60		70	
Compressor Discharge Pressure PSIG												
Compressor Discharge Temp °F			160		290		312		412		485	
Clearance Probes												
#1 Radial									.124	.006"+	.124	.006+
#2									.085	.009	.079	.010
#3									1.00	.0085	.098	.009−
#4									.111	.006	.122	.005−
#1 Axial									—		—	
#2									.073	.024	.082	.021
#3									.068	.024	.073	.021
#4									—		—	
Compressor Inlet Pressure												
#1 "H_2O												
#2												
#3												
7			84		86		104		142		136	
6			84		92		10[illegible]		142		144	
29			82		[illegible]		102		137		138	
30			84		86		92		115		124	
31			82		84		90		112		122	
32			83		76		74		116		128	
33			—		—		—		—		—	
34			—		—		—		—		—	
35			126		156		212		280		302	
36			160		[illegible]		298		297		444	
37			162		300		300		406		454	
38			160		315		294		392		430	
39			90		104		125		156		210	

SEE CALIB Y-317

NOTE- AXIAL CLEAR. CHGD WITH TIME — MAY BE .005" LESS IF UNIT IS BROUGHT UP TO SPEED QUICKLY

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

NASA 6:1 TEST RIG
BUILD NO. 4A

AiResearch Manufacturing Company of Arizona

TABLE XII. (Contd)

72

NASA 6:1 COMPRESSOR RIG

DATE: 8-11-72
OPERATOR: E. [illegible]
ASSISTANT: B. [illegible]

			AM	PM	AM	PM	AM	PM	AM	PM	AM	PM		RUN # 2		
Speed	0		40,000		50,000		60,000	60,000	70,000	70,000	80,000	80,000				
Oil Inlet Pressure PSIG			65		65		65	65	65	65	~~65~~	65				
Oil Inlet Temperature °F			120		120		123	122	125	130	~~122~~	122				
Oil Flow GPM			—													
Compressor Bearing Temperature °F																
#1			155		130		150	145	160	165		170				
#2			155		130		145	150	155	165		180				
#3			155		130		145	150	150	145		180				
Turbine Bearing Temperature °F																
#1			—		—		—	—	—	—		—				
#2			110		130		165	150	175	210		295				
#3			125		130		155	145	155	175		180				
Thrust			75		75		60	75	50	50		160				
#1																
#2																
#3																
Thrust Chamber Pressure PSIG			2		4		6.3	6.0	10	7.2		14				
Vibration (Diff) g's																
Vibration (Housing) g's																
Shaft Excursion																
#1 Turbine																
#2 Turbine																
#1 Compressor																
#2 Compressor																
Turbine Inlet Temperature °F			230		260		290	300	320	360		430				
Turbine Inlet Pressure PSIG			15		30		50	50	80	70		115				
Turbine Discharge Pressure PSIG																
Diffuser Force Lbs			125/125		215/215		335/335	305/305	480/480	415/415		STUCK	[illegible] 350°F [illegible]			
Gas Bearing Pressure (Top) PSIG												575/585				
Gas Bearing Pressure (Btm) PSIG																

ALTITUDE EQUIPMENT DIVISION	NASA 6:1 TEST RIG BUILD NO. 4A
CALCULATED	
RECORDED	
DRAWN	
CHECKED	
APPROVED	AiResearch Manufacturing Company of Arizona

TABLE XII. (Contd)

NASA 6:1 COMPRESSOR RIG

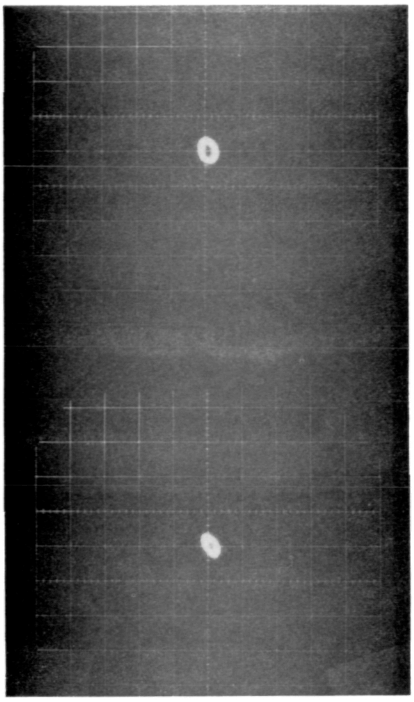
			AM	PM	AM	P	AM	PM	AM	PM	AM	PM				
Speed	0		40,000		50,000		60,000	60,000	70,000	70,000	70,000	80,000				
Gas Bearing Temperature (Btm) °F			55		[illegible]		55	180	60	165		280	(360)			—
Compressor Discharge Pressure PSIG																
Compressor Discharge Temp °F			141		171		264	280	356	410		462				
Clearance Probes																
#1 Radial									.132-.140							
#2									.089-.094							
#3									.094-.100							
#4									.106-.113							
#1 Axial									.073							
#2									.074							
#3									.070							
#4									—							
Compressor Inlet Pressure																
#1 "H_2O																
#2																
#3																
27			70		68		82	86	94	132		116				
28			64		62		74	84	84	128		108				
29			66		64		76	82	90	120		114				
30			60		56		68	70	62	87		82				
31			58		54		65	70	60	84		75				
32			62		58		60	74	65	88		85				
33			—		—		—	—	—	—		—				
34			—		—		—	—	—	—		—				
35			104		[illegible]		[illegible]	174	[illegible]	[illegible]		276				
36			130		172		236	263	338	352		436				
37			136		[illegible]		260	271	355	362		458				
38			134		[illegible]		262	262	324	347		425				
39			[illegible]		96		115	199	153	220		375				

ALTITUDE EQUIPMENT DIVISION			NASA 6:1 TEST RIG BUILD NO. 4A	TABLE XII. (Contd)
CALCULATED				
RECORDED				
DRAWN				
CHECKED				
APPROVED			AiResearch Manufacturing Company of Arizona	

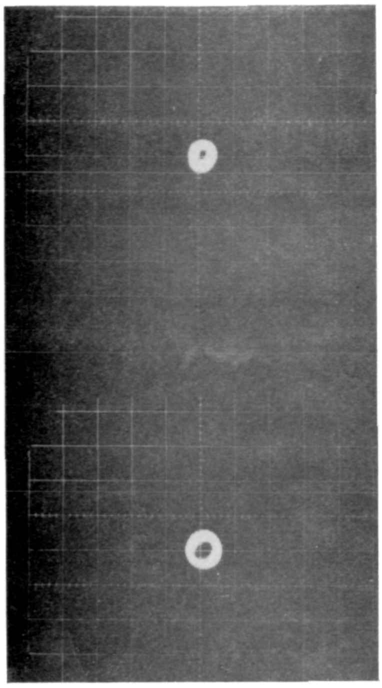
NASA 6:1 COMPRESSOR RIG
BUILD 4A LISSAJOUS TRACES

COMPRESSOR END

TURBINE END



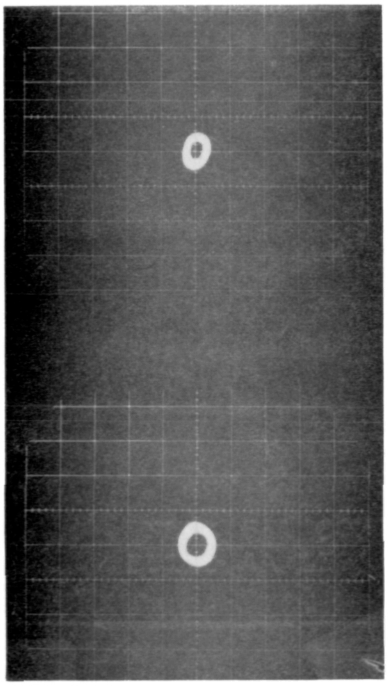
50,800 RPM



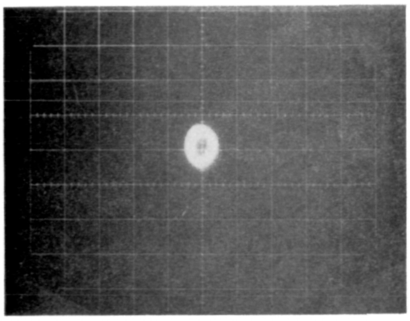
60,250 RPM

COMPRESSOR END

TURBINE END



70,250 RPM



TURBINE PROBES NOT WORKING

80,000 RPM

Figure 44.

SCALES: 1.0 MIL/DIV
0.2 VOLT/DIV
TEST DATE: 10 AUG. 1972

MP-37110

Build 5 and 5A

After the previous rig build of August 11 (designated herein as Build 4A) was removed from the test cell, a meeting was held with the NASA Project Manager during which an agreement was made to check the axial and radial motion of the rotating group inside the housings. This deflection test was desirable since previous rig builds could successfully float the diffuser, i.e., with intermittent electrical continuity, by removing the front journal gas bearing support. This front support was originally designed about the inlet housing to provide stability to the floating shroud/diffuser assembly over and above the support from the thrust gas bearing. However, without the front journal gas bearing support, the possibility of additional motion of the shroud/diffuser assembly gave rise to considering the rub potential in the compressor. Hence, a deflection test could resolve this concern.

The rig was reassembled, designated now as Build 5, without the carbon face seals or preload spring. The rig was oriented in the vertical shaft position with the compressor down. Rotor movement was observed as described in the following table.

Test Condition	Load, lb Toward Comp.	Rotor Movement, in.*	Remarks
1	0	0.007	Axial play of compressor bearing
2	138	0.011	Movement due to deflection of retainer spring closing clearance gap and deflection of bearing retainer

*By dial indicator measurement.

A determination was also made of the maximum change in radial operating clearance between a cocked shroud/diffuser assembly to the compressor inducer as compared with the clearance when the rotor and shroud are centered. This condition resulted in a total clearance decrease of 0.0015 in. Use of feeler gauges to perform radial measurements added some inaccuracy to the test but the significant fact is that, even with a small measurement inaccuracy of about ±0.001 in., the 0.010-in. inducer to shroud radial clearance was sufficient for operation at design speed with a 0.002-in. inducer blade growth and all parts displaced to their closest radial positions. Similarly, a high rotor thrust reversal could permit a 0.011-in. axial rotor motion which corresponds to the axial clearance at design operating speed. At design speed, however, the turbine inlet pressure bleeds compressed air into the thrust piston cavity providing a

positive thrust condition on the rotor which makes thrust reversal abnormal. At lower operating speeds, the compressor axial clearance increases due to a reduction in the axial flowering effect in the wheel from centrifugal stresses. By previous test measurements this effect was determined to be 0.010 in. from zero to design speed. The conclusion from the deflection test was that radial and axial compressor rubs could not occur with the Build 5 configuration.

The journal gas bearing support was added to Build 5, and air pressure was supplied to the floating diffuser. Electrical continuity through a Simpson meter (Rx100 scale) was intermittent and sensitive to loading the floating diffuser on one side. The journal gas bearing support was removed, and the electrical condition was unchanged. The floating diffuser was positioned to the condition of maximum electrical continuity, and a 2000-microfarad capacitor charged at 40 v was discharged through the parts several times. This power source was sufficient to raise metal at the interfering metal parts. Build 5 was disassembled and markings were noticeable between a chamfered face on the backside of the shroud/diffuser assembly and a chamfered face on the inside of the compressor collector housing, SKP25673-1. The thrust gas bearing was free of electrical markings from the capacitor discharge. Remachining of the chamfer inside the compressor collector housing was instituted to provide an additional 0.010-0.015 in. radial clearance for the floating diffuser assembly. All parts comprising the thrust gas bearing were inspected including the axial passage width of the assembled air bearing housing, SKP25663-1 and air bearing ring, SKP25662-1. These parts were to design requirements which resulted in 0.007-in. axial clearance and 0.005-in. diametral clearance in the bearing.

Build 5A was assembled using a new inducer with radial hand finish work on the blades, oil seal elements with antirotation features, the reworked compressor case and clearance spacers, and the front gas journal bearing support. Electrical continuity was not measurable through the floating diffuser except for intermittent low values which appear to be constantly varying, probably due to the air supply cleanliness (with a 5-micron air filter passing 0.0006-in. maximum particle sizes through a 0.001-in. air gap) or the felt metal seal fibers contacting the labyrinth knife edges. Assembly data for this build is given in Table XIII.

The test rig was operated to 51,000 rpm in the test cell where three problem areas were identified. First, at 51,000 rpm the axial clearance probes were reading 0.002- to 0.003-in. clearances which restricted further speed increases. This was inconsistent with previous build measurements and unacceptable for shipping. Radial probes, however, read 0.007- to 0.010-in. clearances which were deemed satisfactory. Second, the turbine oil seal would permit oil leakage

TABLE XIII.

BUILD 5A DATA SHEET
NASA 6:1 COMPRESSOR

A. Runout

1. Turbine

a. O.D.	0.0003
b. Front Face	0.0008
c. Knife Edge Seal	0.0003

2. Compressor

a. O.D.	0.0007
b. Back Face	0.0008
c. Knife Edge Seal	0.0006

B. Balance

Balance	Max Allowed	Actual
1. Turbine	0.017 Oz-In.	0.016
2. Compressor	0.023 Oz.-In.	0.005

C. Clearances

Clearances	B/P	Actual
1. Turbine Bearing Housing to Seal Retainer	0.023-0.027	0.027
2. Compressor Bearing Housing to Seal Retainer	0.001-0.003	-0.001
3. Diffuser Knives to Discharge, Turbine End	0.003-0.005	0.010
4. Diffuser Knives to Discharge, Compressor End	0.003-0.005	0.014 - 0.016
5. Turbine Wheel Clearance	0.023-0.027	0.058
6. Compressor Face Clearance	0.021-0.023	0.022

when thrust cavity pressure exceeded 15 psig at speeds exceeding 30,000 rpm. This also was inconsistent with the previous build (Build 4A) in which 60 psig thrust cavity pressures would seat the secondary Teflon ring seal on the turbine end but pressures of 75 psig now failed to do so. Third, a vibrational peak of 13 g's at 51,000 rpm was being recorded which was not previously predicted from interference diagrams but had been observed on a previous test. This vibrational peak is shown in figure 45 with the corresponding shaft lissajous trace in figure 44. Other test data appears in Table XIV.

An AiResearch review board, consisting of engineering and laboratory personnel, was called to emphasize, demonstrate, and resolve the Build 5A problem areas. The resolutions from these efforts were as follows:

(a) The capacitance probe readout instrumentation was properly functioning, and the inconsistency in axial clearance data would require recalibration and closer inspection of the disassembled compressor parts to confirm the correct spacer thicknesses.

(b) The vibrational readout data from the accelerometer was originating from a housing natural frequency since no lissajous reading increases were observable from the Bentley probes. The amplitude of vibration at the 51,000 rpm speed was calculated to be 0.00025 in. and within reasonable limits.

The compressor rig was removed from the test facility.

Build 6

The compressor rig was disassembled to determine the causes of the problem areas identified during Build 5A testing.

Axial clearance probe readings were confirmed as correct by a recalibration of the shroud and impeller in the Instrumentation Laboratory. Inspection of parts confirmed the instrumentation readings received. A review of clearance data from previous tests indicated a history of clearance data as shown in Table XV. A set of inconsistent measured axial clearance appears for Build 4A and, although no obvious explanation exists, it is most likely the result of oil and dirt contamination on the probes. Additionally, the 0.008-in. recession of the probes beneath the shroud contoured surface was entered on the table to avoid interpretation difficulties during future inspections.

The bearing and seal cavity was disassembled and seal rotors and carbon elements were checked for flatness and runout. Both parameters

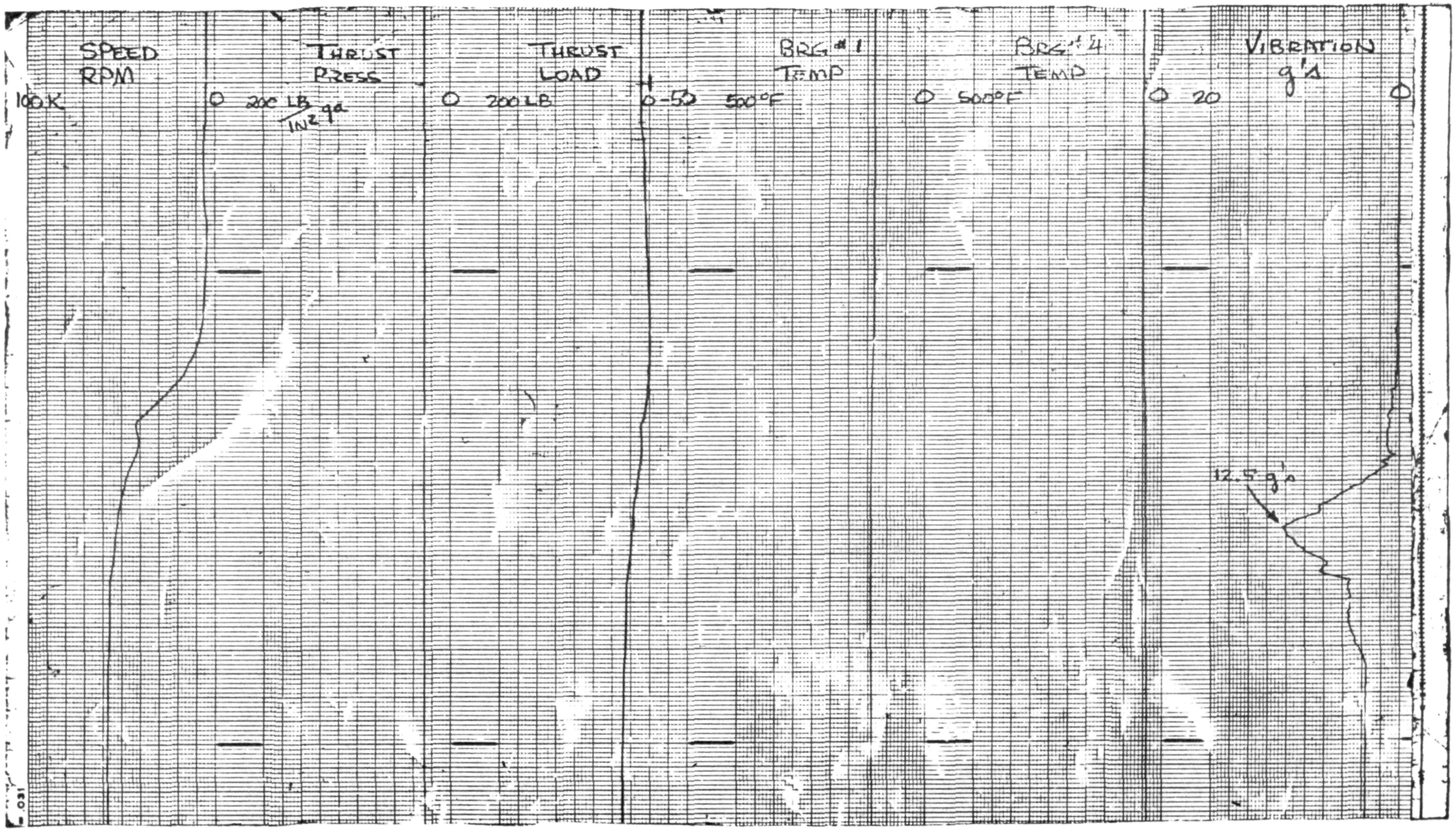


Figure 45. Build 5A Test Data NASA 6:1 Compressor

NASA 6:1 COMPRESSOR RIG

DATE: ________
OPERATOR: ________
ASSISTANT: ________

Speed	6880		2000													
Oil Inlet Pressure PSIG 16 [illegible]	[illegible]															
Oil Inlet Temperature °F 10/25	[illegible]															
Oil Flow GPM CP/GPM	[illegible]															
Compressor Bearing Temperature °F																
#1 CH 1	117															
#2 2	115															
#3 3	115															
Turbine Bearing Temperature °F																
#1 CH 4	114															
#2 5	110															
#3 6	110															
Thrust																
#1																
#2																
#3																
Thrust Chamber Pressure PSIG	20															
Vibration (Diff) g's	12															
Vibration (Housing) g's																
Shaft Excursion																
#1 Turbine																
#2 Turbine																
#1 Compressor																
#2 Compressor																
Turbine Inlet Temperature °F CH 7	[illegible]															
Turbine Inlet Pressure PSIG [illegible]	[illegible]															
Turbine Discharge Pressure PSIG																
DIFF FORCE LBS	-15.8															
Diffuser Force Lbs	[illegible]															
Gas Bearing Pressure (Top) PSIG	[illegible]															
Gas Bearing Pressure (Btm) PSIG	[illegible]															

ALTITUDE EQUIPMENT DIVISION		
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APPROVED		

BUILD 5A TEST DATA
OCTOBER 4, 1972

AiResearch Manufacturing Company of Arizona

TABLE XIV.

NASA 6:1 COMPRESSOR RIG

CP

Speed		6800	~~42000~~	4000		55000	[illegible]
Gas Bearing Temperature (Btm) ^{o}F					[illegible]	71	70
					[illegible]	72	70
Compressor Discharge Pressure PSIG		[illegible]					
Compressor Discharge Temp ^{o}F		[illegible]					
BALANCE PISTON INLET AIR TEMP							
Clearance Probes							
#1 Radial		.119	.007"	.110	.0071		
#2		.099	.0105"	.070	.012		
#3		.109	.0075"	.094	.0072		
#4		.094	.008	.085	.010		
#1 Axial	9	—	—	—	—		
#2	10	.222/.214	.002/.002	.154	.008		
#3	11	.167	.003/	.172	.0081		
#4	12	—	—	—			
Compressor Inlet Pressure							
#1	"H_2O						
#2							
#3							
	27	94					
COMPR INLET DUCT SKIN	28	100					
	29	92					
	30	90					
JOURNAL AIR BRG SKIN	31	91					
	32	91					
	33	196					
SHROUD SKIN	34	196					
	35	190					
	36	234					
DIFF SKIN	37	233					
	38	232					
COMPR INLET	39	93					
" "	40	96					

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED		

BUILD 5A TEST DATA
OCTOBER 4, 1972

AiResearch Manufacturing Company of Arizona

TABLE XIV. (Contd)

TABLE XV.

SUMMARY OF CLEARANCE MEASUREMENTS
COMPRESSOR TEST RIG

Clearance Description	Build Number 3	4	4A	5A	6
Build Axial Clearance - Probe to Impeller	0.058	0.028	0.032	0.022	0.032
Build Axial Clearance - Shroud to Impeller	0.050	0.020	0.024	0.014	0.024
Axial Running Clearance	0.051 @ 4000 rpm 0.045 @ 50,000 rpm 0.041 @ 80,000 rpm	0.0155 @ 32,000 rpm 0.003 @ 32,000 rpm	0.025/0.026 @ 70,000 rpm 0.022/0.023 @ 80,000 rpm	0.006 @ 4000 rpm 0.002 @ 50,000 rpm	0.0134 @ 80,000 rpm
Running Clearance Change For Next Build	-0.031	-0.005	-0.010	+0.010	-
Spacer Thickness (Large)	0.1600 0.1605	0.1500 0.1505	0.1550 0.1555	0.1450 0.1455	0.1550 0.1555
Spacer Thickness (Small)	0.1350 0.1355	0.1250 0.1255	0.1300 0.1305	0.1200 0.1205	0.1300 0.1305
Diffuser Ring Clearance Change	-0.020	-	-	-	-
Design Running Clearance	0.010 ±0.001 @ 80,000 rpm				
Design Axial "Flowering Effect"	0.012 @ 96,000 rpm 0.010 @ 80,000 rpm				
Build Clearance Required	0.022 ±0.001				

NOTES:
1. All clearance measurements in inch units
2. Clearance data reflect average values

were found to be satisfactory. Stacking of seals, springs, anti-rotation spacers, and retainers was checked which revealed that the compressor seal was adequately stacked and the turbine seal spring load could be increased by an additional 0.030-in. compression to reduce the susceptibility to oil leakage at high speeds. The carbon elements showed no signs of outside diameter wear after the anti-rotation spacers were added prior to Build 5A. The secondary teflon ring seals showed no indications of damage but a preference was expressed by the customer and the AiResearch Seals Engineering Department to use alternate silicone O-rings for the next build.

In order to have some assurance that the seals would be adequate during the next test, a static pressure test was devised to check seal leakage using the bearings and seals assembled into their support housing and with the compressor and turbine wheels assembled on the shaft. The turbine plenum was also used in this assembly such that a 300-psig remote air supply could be fed to the thrust balance piston. Pressure gauges were added to the thrust balancing cavity and the oil scavenging cavity and observed. While pressure in the thrust balancing cavity was raised from 0 to 90 psig (90 psig was the limiting level achievable from this source since the thrust cavity bleeds off to ambient), a small vacuum pump maintained the oil scavenge cavity at a constant condition of 22 in. of mercury vacuum (-22 in. Hg gage). Rotation of the rotor by hand at several thrust pressure levels did not affect the scavenge cavity pressure so the assembly was accepted as the beginning of Build 6.

Build 6 was completed using the new inducer with radial hand finish work on the blades, oil seal elements with anti-rotation features, and the front journal gas bearing support as used on Build 5A. Secondary seals were changed from teflon rings to silicone O-rings and the turbine seal spring load increased as noted in the preceding discussion. Spacer rings were increased 0.010 in. to the same size as those used on Build 4A which would give axial running clearances at full speed of about 0.013-0.014 in. Careful assembly was practiced to eliminate lockwire contact with the floating diffuser assembly, cleanliness of the gas bearing assembly, and inspection of axially stacked parts. Confirmation of the 0.024-in. axial clearance was made in the completed assembly by using the calibration instrumentation. Build parameters for this assembly are given in Table XVI.

Installation of the rig in the test cell was completed and operation to 80,000 rpm was performed. Around 70,000 rpm, the rig began leaking oil. Vacuum gages were connected to the turbine and compressor scavenge lines to observe the nature of the oil leakage conditions. Data was taken at various speeds using one, two, or three gear-type scavenge pumps connected in parallel to the bearing and seal cavity. Results of these tests are shown in Table XVII. These data indicated that the compressor end scavenge vacuum was lost with speeds over

TABLE XVI.

BUILD 6 DATA SHEET
NASA 6:1 COMPRESSOR

A. Runout

1. Turbine

a.	O.D.	0.0003
b.	Front Face	0.0003
c.	Knife Edge Seal	0.0003

2. Compressor

a.	O.D.	0.0007
b.	Back Face	0.0008
c.	Knife Edge Seal	0.0006

B. Balance

		Max Allowed	Actual
1.	Turbine	0.017 Oz-In.	0.016
2.	Compressor	0.023 Oz.-In.	0.005

C. Clearances

		B/P	Actual
1.	Turbine Bearing Housing to Seal Retainer	0.023-0.027	0.028
2.	Compressor Bearing Housing to Seal Retainer	0.001-0.003	0.001
3.	Diffuser Knives to Discharge, Turbine End	0.003-0.005	0.010
4.	Diffuser Knives to Discharge, Compressor End	0.003-0.005	0.014 - 0.016
5.	Turbine Wheel Clearance	0.023-0.027	0.058
6.	Compressor Face Clearance (Shroud to Impeller)	0.021-0.023	0.024

TABLE XVII.

OIL CAVITY-SCAVENGE PUMP STUDY
BUILD 6, NASA 6:1 COMPRESSOR RIG

Number Scavenge Pumps	1		2				3	
Rotor Speed, rpm	0 (Development Assembly Pump)	0 (Test)	0	4K	65K	70K	0	70K
Compressor Scavenge Pressure, in. Hg-Ga	-22	-18	-22	-20	-6	-4	-20.75	-19.5
Turbine Scavenge Pressure, in. Hg-Ga	-22	-22	-22	-25	-15	-15	-24	-24
Thrust Pressure, psig	0	0	0	0	0	56.7	0	60
Thrust Load, lb	0	0	0	0	0	170	0	180

70,000 rpm when using a two-scavenge pump system. Using a three-scavenge pump system, the rig could be operated to 80,000 rpm without oil leakage; but, after about 10 min of operation, the compressor end scavenge vacuum was again lost. The following observations resulted from these tests.

(a) The addition of the extra scavenge capacity of a third gear-type pump was a significant improvement in the operating range of the compressor rig seals without oil leakage.

(b) The interfacing lubrication and scavenge system would largely influence the successful operation and acceptance of the rig. Seal leakage would occur whenever the compressor (or turbine) scavenge pressure rose above -5 in. mercury gage (12.2 psia).

(c) The scavenge vacuum could also be restored to the seal cavity if a thrust pressure was added to the balance piston cavity. The balance piston cavity bleeds off to ambient via an intermediate pressure which is vented through six holes on the turbine side of the turbine end seal. Although this intermediate pressure is very low, it could be raised by plugging some of the holes to provide a higher Δp across the turbine seal if desired. This approach would be considered secondary to the correct matching of a lubrication/scavenging system.

(d) Since the specific seal responsible for oil leakage could not be positively identified, the possibility of a compressor seal problem was considered. However, the compressor seal Δp is increased when the compressor is operated at discharge pressures other than the choke conditions of the previous tests which would be favorable for maintaining scavenge vacuum.

Axial and radial clearances were read during the above checkout tests. Axial measurements were 0.012 and 0.013 in. at 80,000 rpm. Two radial clearance probes at 90 degrees were reading about 0.002 in. whereas two others were reading 0.010 in. at 70,000 rpm. Radial readings were inconsistent with those taken in Build 5A and measured hardware. The rig was shut down, measurements checked, probes washed with a cleaning fluid, and torque rings adjusted. No significant change could be made to the radial clearance probe readings.

The rig was returned to the Development Assembly area to investigate the radial probe reading inconsistency. Calibration instrumentation was used to provide readouts while the rotor was rotated and the shroud moved radially across the gas journal bearing clearance space. Radial deflections of 0.001 to 0.0046 in. were imposed and measured. Two probes could not, however, read radial clearances as measured using feeler gages and shim stock and were therefore concluded to be unreliable, probably due to oil and dirt contamination.

The rig was returned to test as Build 6A and the 5-hr mechanical acceptance test was begun. The compressor was operated near choke conditions at 80,000 rpm with data recorded as shown in Table XVIII. Thrust load was held at 340 lb instead of increasing the scavenge capacity of the interfacing pumps. Lissajous plot from the Bentley probes showing rotor excursion at design speed is shown as figure 46. Starting, operating, and shutdown performance is documented on recording charts shown as figures 47 and 48.

The research package was delivered to the NASA Lewis Research Center in Cleveland, Ohio.

An analysis of the influence of a 270 lb aerodynamic thrust load on the average bearing fatigue life was performed after delivery of the research package. Figure 49 shows the average bearing B_4 fatigue life before and after the 5-hour mechanical test. Average life, as represented by 96 percent of a group of bearings, is plotted versus the aerodynamic thrust loading measured axially on a thrust ring zero calibration represented a 70 lb axial preload imposed by the rotor weight and a thrust reversal spring force. The aerodynamic thrust load of 270 lb therefore represented a total thrust bearing load of 340 lb when the bearing preload is added. For a 300 hr design life, figure 49 indicates an average aerodynamic thrust load of 162 lb should be maintained if all of the 300 hour average bearing life were to be expended at full speed of 80,000 rpm and under a 53 lb dynamic radial load. After five hours operation at 340 lb total thrust load, a remaining bearing design life of 295 hours would be achieved if the average aerodynamic thrust on future testing is limited to 160 lb. Again this assumes expending all of the bearing fatigue life at 80,000 rpm and under a 53 lb dynamic radial load.

The radial bearing loads for the fatigue life calculations are based on an AiResearch design procedure which considers the following rotor dynamics analysis:

The dynamic response of any rotor system to rotor unbalance is a function of the angular phase relation between rotor component center of gravity eccentricities. In fact the magnitude of the response can be a very strong function of those phase relationships in some cases (e.g. near a node in the rotor). The dynamic bearing loads and therefore, the predicted bearing life will then be a function of the eccentricity phase relation of the individual components. It may be noted that this dependence of rotor response magnitude on eccentricity phase relation is a result of the dynamic deflection that takes place in any rotor operating near or above any of its critical speeds.

A convenient analytical eccentricity distribution of a rotor is to assume all eccentricities are in phase (in-phase or normal bearing loads). Additionally, the analytical method should calculate the bearing loads for the worst possible eccentricity phase relationship

FORM P-766

10-11-72
CA-2
WHITTEN
STEWART
PENNETT

NASA 6:1 COMPRESSOR RIG

TIME OF DAY, HOURS					0655	0725	0755	0855	0955	1055	1155	
Speed	0	4000	40,000	64,400	80,000	80,000	80,000	80,000	80,000	80,000	80,000	* ~~96,000~~
Gas Bearing Temperature (Btm) °F												
Compressor Discharge Pressure, IN HG GA. (WET 1573)		-θ-	7.5/4.5	21.0/15.0	320/32.0	[illegible]/32.3	34.0/32.5	34.0/32.5	34.4/32.6	34.3/32.5	34.2/32.3	✓
Compressor Discharge Temp °F TC 26		-	144	324	405	50[illegible]	505	510	513	512	516	✓
BALANCE PISTON INLET AIR TEMP CA 27		65	65	65	61	68	70	78	77	80	70	
Clearance Probes												
#1 Radial 1		.006	.005	.0035	.002	.003	.0031	.0031	.0031			✓
#2 2		.0025	.0039	.0035	.001	.001	.001	.001	.001			✓
#3 3		.0105	.0104	.010	.0098	.008	.008	.008	.008			✓
#4 4		.012	.010	.0085	.0088	.0085	.0085	.0085	.0085			✓
#1 Axial 9		—	—	—	—	—	—	—	—	—		✓
#2 10		.024	.0202	.016	.014	.0135	.0135	.0135	.0135			✓
#3 11		.024	.020	.0155	.014	.014	.0138	.0138	.0138			✓
#4 12		—	—	—	—	—	—	—	—	—		✓
SCAVENGE Pressure												
#1 "Hg (2) TURB SCAV		-20.75	-20.75	-13.0	-15.0	-1[illegible].0	-18.0	-17.5	-17.7	-17.7	-17.7	
#2 1 COMPR. SCAV		-24.0	-23.5	-22.5	-22.7	-23.0	-23.0	-22.5	-22.7	-22.5	-22.5	
#3												
27		60	73	110	159	154	152	155	158	157	160	
COMPR INLET DUCT - SKIN 28		61	70	104	135	134	134	139	140	140	142	
29		61	71	106	134	131	134	135	135	135	137	
30		56	64	93	118	117	118	124	122	122	124	
JOURNAL AIR BRG - SKIN 31		56	64	86	120	124	124	126	128	129	130	
32		58	63	80	100	98	100	105	105	105	106	
33		63	129	270	396	398	396	401	405	404	405	
SHROUD - SKIN 34		63	126	268	402	406	404	408	410	410	414	
35		63	123	266	390	380	390	388	388	388	390	
36		63	155	336	478	480	440	485	488	488	490	
DIFFUSER - SKIN 37		63	155	340	474	490	478	485	496	486	490	
38		63	158	342	477	492	482	486	489	487	492	
COMPRESSOR INLET AIR 39		61	73	122	136	130	131	134	135	135	137	
40		61	63	100	80	72	71	74	76	78	80	

* DO NOT EXCEED 5 MINUTES TOTAL OPERATION AT THIS SPEED (ONLY ✓ DATA)

ALTITUDE EQUIPMENT DIVISION	
CALCULATED	
RECORDED	
DRAWN	
CHECKED	
APPROVED	Sh[illegible]

NASA 6:1 COMPRESSOR RIG, BUILD NO. 6 5 HOUR TEST

TABLE XVIII.

AiResearch Manufacturing Company of Arizona

NASA 6:1 COMPRESSOR RIG

DATE: 10-11-72
OPERATOR: WHITTEN - STEWART
ASSISTANT: BENNETT
CH-2

					0655	0725	0755	0855	0955	1055	1155
Speed	0	4000	40,000	64,400	80,000	80,000	80,000	80,000	80,000	80,000	80,000
Oil Inlet Pressure PSIG LG 30556		75	74	73	70	70	70	70	70	70	70
Oil Inlet Temperature °F IC 25		105	108	138	160	114	106	106	111	113	116
Oil Flow GPM FM-14 CPS/GPM		231/.715	237/.735	349/	336/	297/.925	292/.900	292/.902	299/.925	297/.925	303/.927
Compressor Bearing Temperature °F											
#1 CH 1		100	127	165	196	162	160	158	162	162	165
#2 2		102	127	164	195	158	155	153	157	159	160
#3 3		101	126	162	192	155	152	152	155	156	157
Turbine Bearing Temperature °F		—									
#1 CH 4		100	125	158	182	146	145	145	152	153	155
#2 5		—	—	—	—	—	—	—			
#3 6		100	120	155	181	145	142	145	147	149	150
Thrust											
#1											
#2											
#3											
Thrust Chamber Pressure PSIG LG 5711 NATON		15	30	45	62	66	55	57	61	62.	62
Vibration (Diff) g's		2	3	5	4.5	3.0	4.0	4.0	3.0	3.0	3.0
Vibration (Housing) g's											
Shaft Excursion											
#1 Turbine											
#2 Turbine											
#1 Compressor											
#2 Compressor											
Turbine Inlet Temperature °F CH 7		73	105	363	380	345	343	367	355	360	370
Turbine Inlet Pressure PSIG LG 582		5	20	52	102	110	110	109	111	109	109
Turbine Discharge Pressure PSIG	—	[illegible]	—	—	—	—	—	—	—	—	—
Diffuser Force Lbs X1/X2 T2/T1		-.3/+.3	-4.2/+0.2	-10.7/-.4	-14.2/-0.4	-18.4/-0.7	-17.1/-0.7	-14.6/-0.9	-19.1/-.8	-17.7/-.8	-18.5/-.6
Gas Bearing Pressure (Top) PSIG 341		85	83	82	87	87	85	85	84	84	85
Gas Bearing Pressure (Btm) PSIG 342		127	129	132	134	132	[illegible]	134	132	134	134

ALTITUDE EQUIPMENT DIVISION		
CALCULATED		
RECORDED		
DRAWN		
CHECKED		
APPROVED	Sharp	

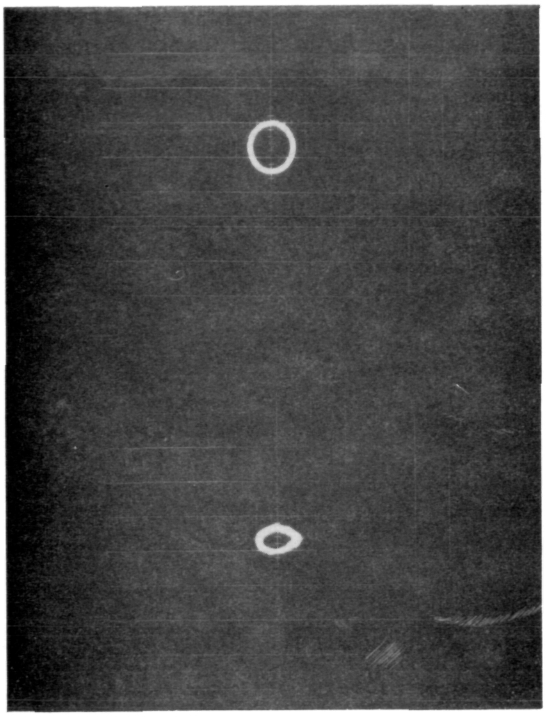
NASA 6:1 COMPRESSOR RIG, BUILD NO. 6 5 HOUR TEST

AiResearch Manufacturing Company of Arizona

TABLE XVIII. (Contd)

NASA 6:1 COMPRESSOR RIG
BUILD 6A LISSAJOUS TRACE

COMPRESSOR
END



TURBINE
END

80,000 RPM

x, y SCALES: 1.0 MIL/DIV
0.2 VOLT/DIV

Figure 46.

MP-37111

Figure 47. NASA 6:1 Compressor 5-Hour Endurance Run at 80,000 RPM Build 6, 10/11/72

PAGE MISSING FROM AVAILABLE VERSION

Figure 48. NASA 6:1 Compressor 5-Hour Endurance Run at 80,000 RPM Build 6, 10/11/72

PAGE MISSING FROM AVAILABLE VERSION

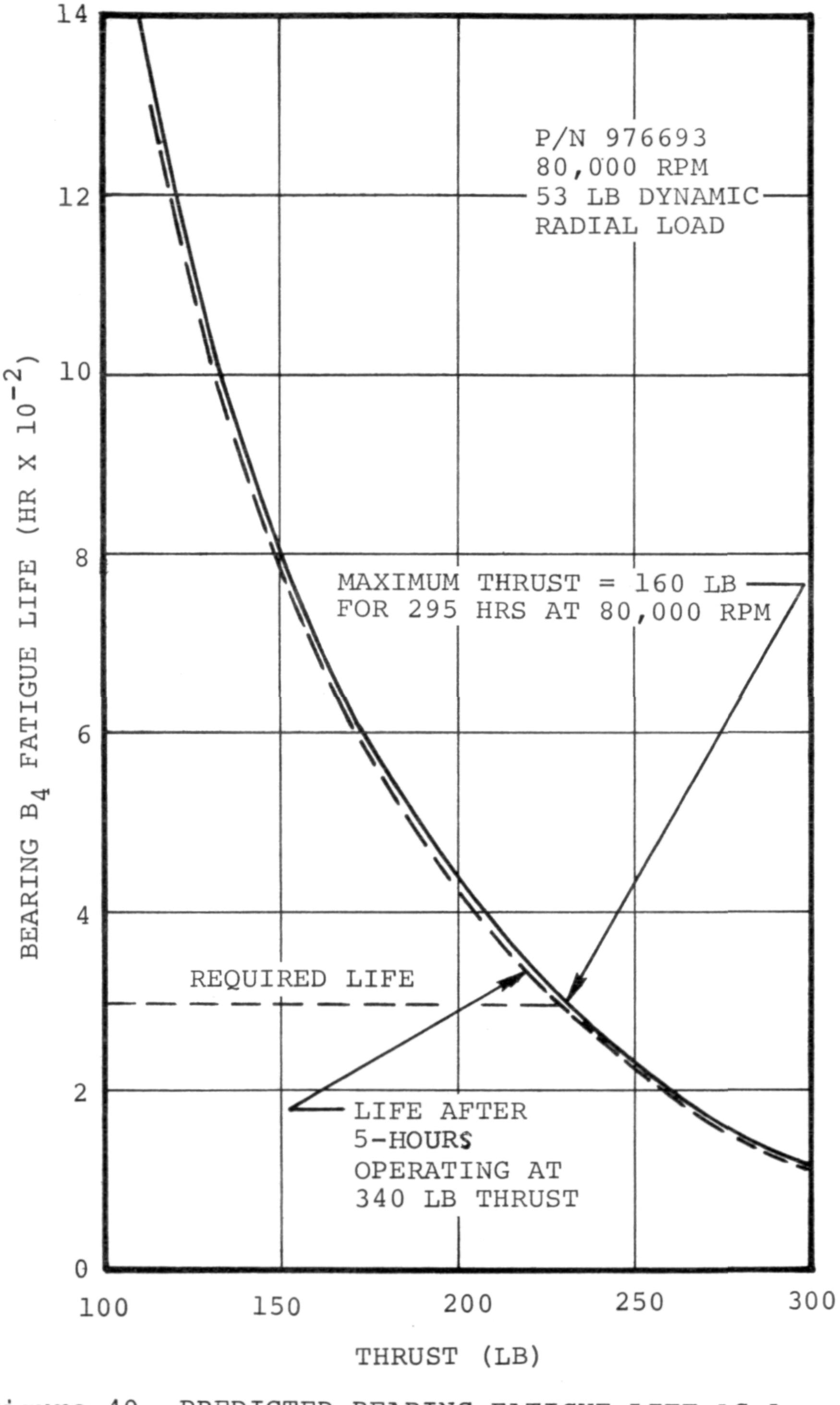


Figure 49. PREDICTED BEARING FATIGUE LIFE AS A FUNCTION OF THRUST LOAD

or worst case bearing loads (also called absolute bearing loads). In some cases though, this worst case method does not give realistic statistical bearing lives. Therefore, the design approach used for bearing life calculations is intended to select a realistic load for the statistical bearing life calculation by using a mean bearing load based on the average of the worst case and the in-phase bearing loads. Figure 50 shows the radial bearing loads resulting from a rotor dynamics analysis based on this analytical method.

The actual rotor may have imbalance and associated bearing loads other than that described by the mean bearing load calculation due to the accuracy of component balancing, component eccentricity phase relationships, and accuracy of the rotor assembly balance. The rotor balance is accomplished by first balancing the impeller and turbine wheel as components and then selectively assembling and checking the complete rotor. The maximum allowable rotor imbalance permitted during balancing is defined for a 0.00035 in c.g. eccentricity as used in the bearing load calculations. Table XVI shows the maximum imbalance permitted and the imbalance measured in the Build 6 rotor. Since the actual imbalance is below the maximum allowed the dynamic bearing loads are probably below the predicted bearing loads.

Prediction of remaining bearing B_4 fatigue life is therefore not easily definable since statistical sampling of life data, component assembly orientation and unbalance, and variable speed test conditions can additionally influence any estimate. Remaining B_4 fatigue life shown in figure 49 therefore represents a correction to the life of an average bearing using present AiResearch design procedures.

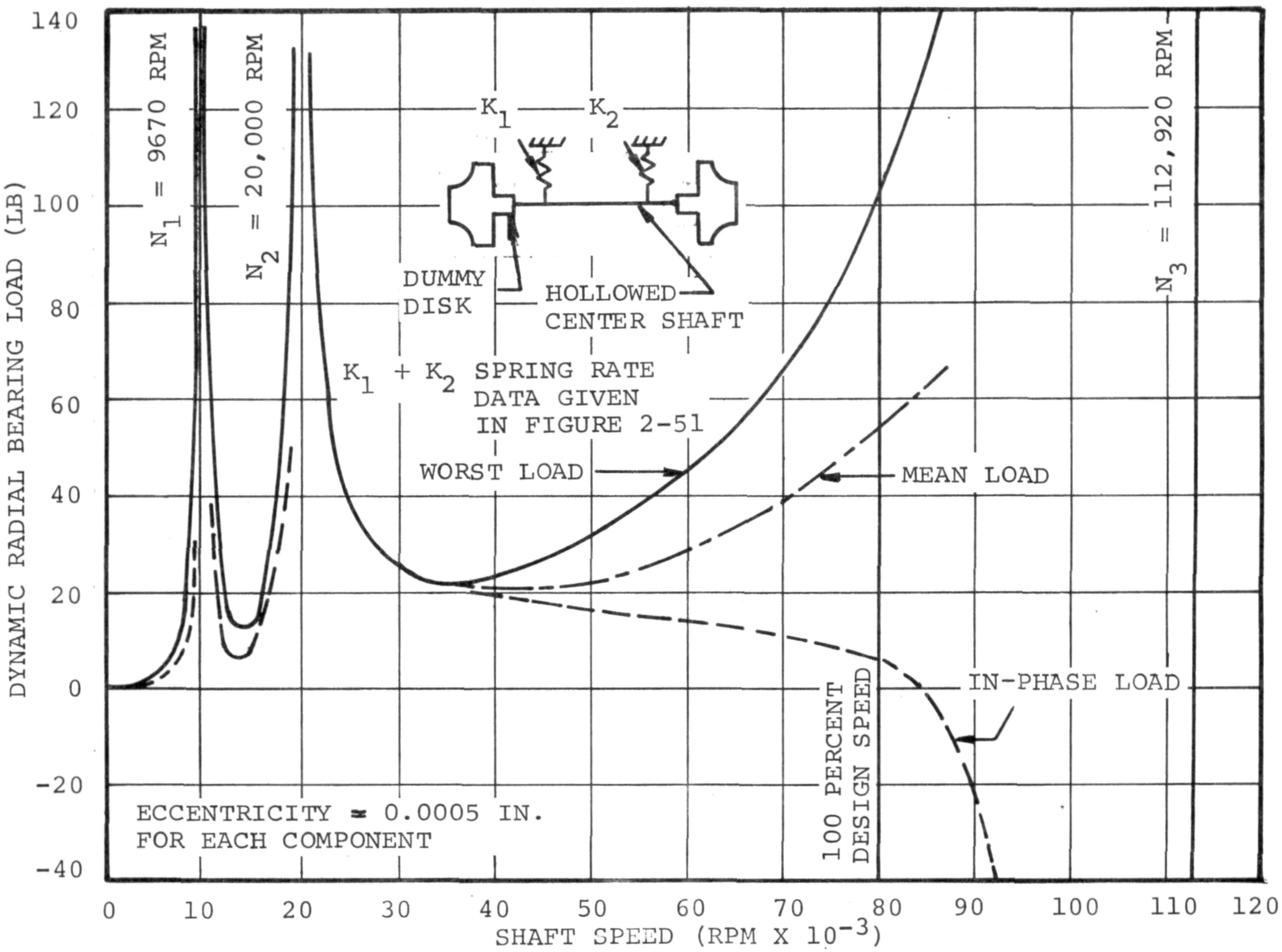


Figure 50. Dynamic Radial Bearing Load Versus Shaft Speed.

APPENDIX I

FAILURE ANALYSIS
OF COMPRESSOR INDUCER
SKP25657-1

(6 Pages)

APPENDIX I

2. FAILURE ANALYSIS OF COMPRESSOR INDUCER SKP25657-1

As discussed under Build 2, the test rig exhibited two modes of distress during operation: (1) failure of two inducer blades, and (2) large excursions of the shaft during operation.

Review of the failed inducer, SKP25657-1, showed that the blades failed in fatigue. A poor surface condition on the blades significantly lowered the fatigue strength of the titanium. A metallurgical examination was conducted to verify the failure mode.

Microexamination showed the fracture surfaces of the two blades to be primarily fatigue and revealed an additional cracked blade (figures 1 and 2). In figure 2, a region of the cracks just above the blade root, can be seen. The trailing edge was apparently thinned and deep scratches can also be seen in this region. Further examination of the blade surface showed scratches over the total pressure and suction surfaces.

The cracked blade was removed and nickel plated so that the edge would not round during polishing for microexamination. The conditions revealed on this blade and on one of the failed blades are shown in figures 3 and 4. Figure 3 is a section perpendicular to the axis of the wheel located 0.010 inch into the trailing edge radius. It shows a deep surface tear at a location near the crack and a possible second tear adjacent to the crack. Figure 4 shows the torn surface as viewed in the same plane as figure 3 and also in a plane perpendicular to the radial direction, both at higher magnification.

Examination of the fracture surfaces indicated that more that one fatigue source was operating. The cracked blade confirms that one major source is on the suction side near the trailing edge radius. The surface examination shows that the axial scratches are deep enough to provide initiation notches for a fatigue crack. The scratches appear particularly severe at the trailing edge just above the blade root. In addition, the trailing edge appears thinner at this point, measuring less than 0.015 inch, contributing to a reduced load carrying capability.

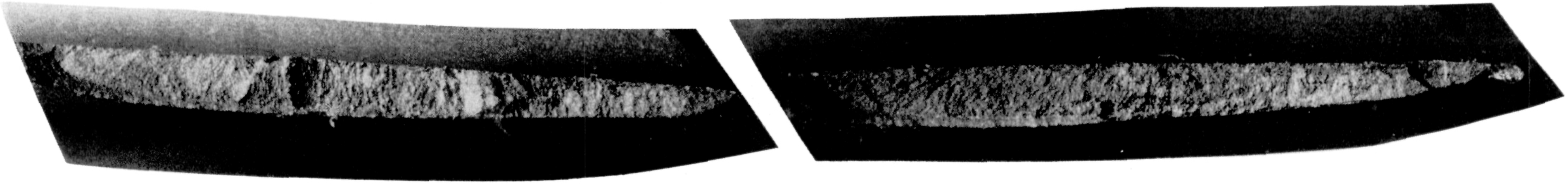
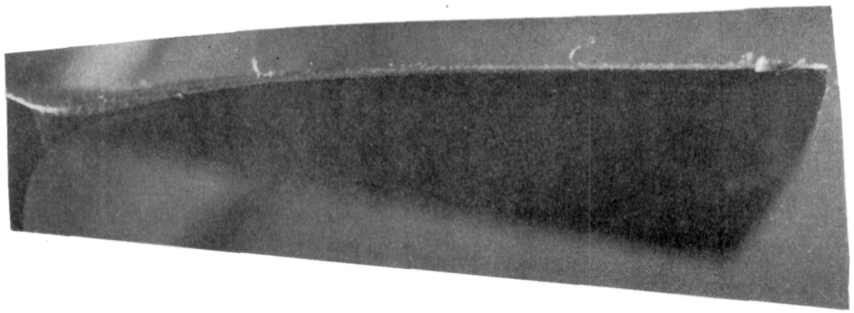
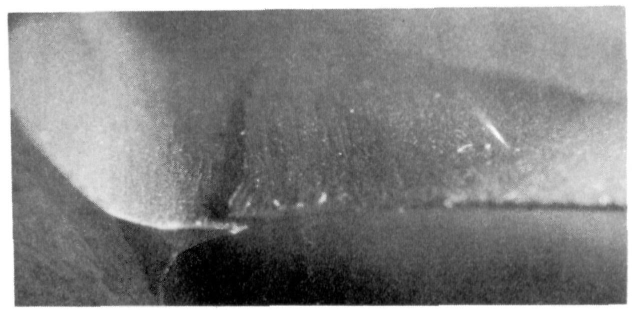


FIGURE 1 - FRACTURE SURFACES OF BOTH FAILED BLADES. CRACK PROGRESSION WAS IN FATIGUE OVER 85% OF THE SURFACE. MAGNIFICATION: 9X



MAGNIFICATION: 5X



MAGNIFICATION 9X

FIGURE 2 - TRAILING EDGES OF TWO BLADES SHOWING THINNING OF FILLET. TYPICAL OF ALL BLADES (LEFT) AND DEEP SCRATCHES TYPICAL OF MANY BLADES (RIGHT).

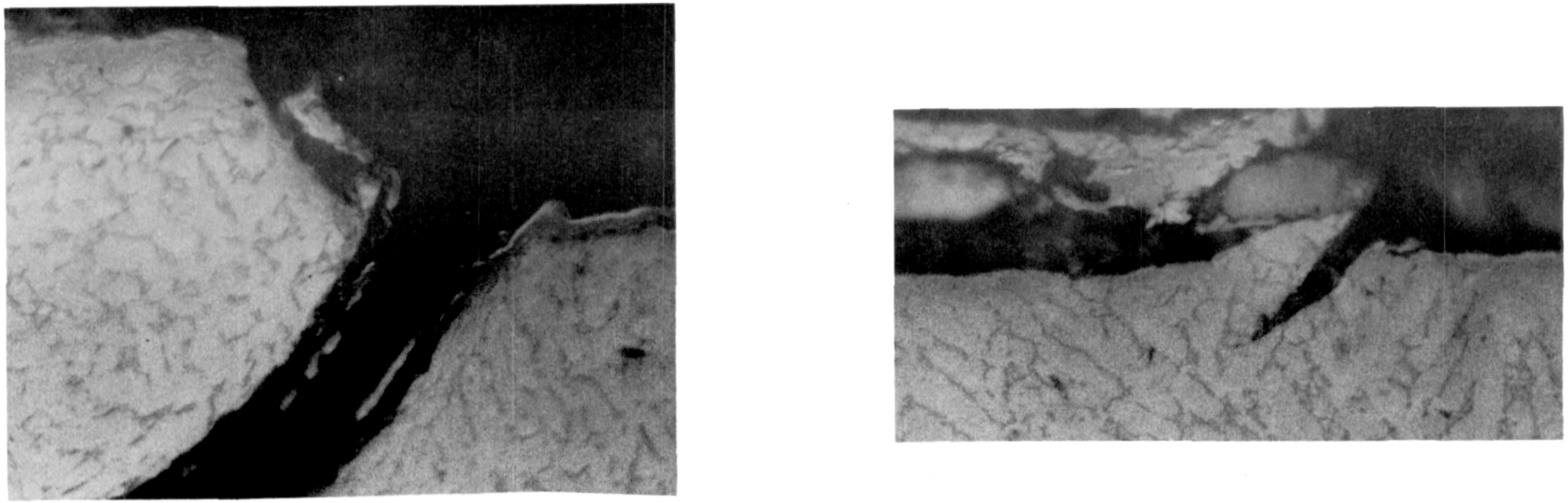


FIGURE 3 - SUCTION SURFACE OF VANE AT TRAILING EDGE. PHOTO ON RIGHT IS JUST ABOVE CRACK (UNETCHED AREAS ARE NICKEL PLATE USED TO RETAIN SPECIMEN EDGE). PLANE OF SECTION PERPENDICULAR TO AXIS OF WHEEL.

ETCHED MAGNIFICATION: 500X

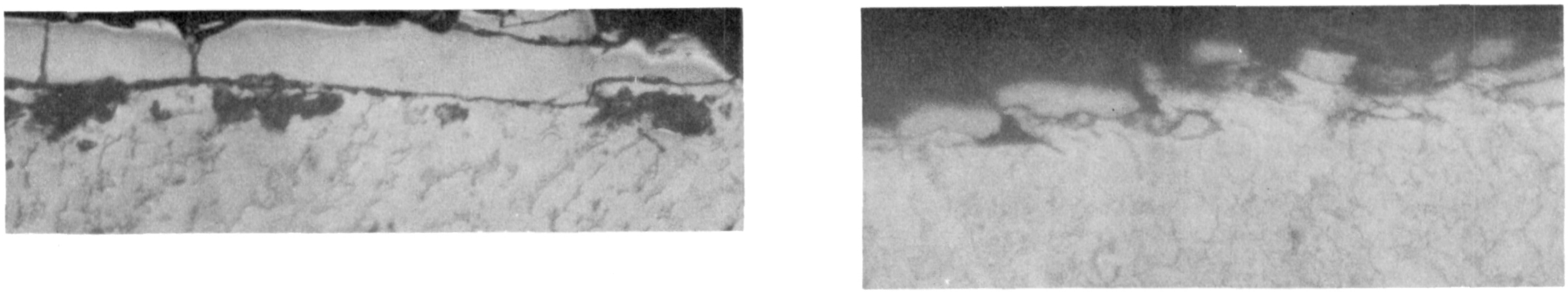


FIGURE 4 - TYPICAL MACHINED SURFACES OF BLADES. SECTION ON LEFT PERPENDICULAR TO RADIAL DIRECTION, ON RIGHT PERPENDICULAR TO AXIAL DIRECTION. LOCATION JUST ABOVE BLADE ROOT ON PRESSURE SURFACE OF BLADE.

ETCHED MAGNIFICATION: 500X

The microexamination revealed major tears and a disturbed metal surface from machining. These also provide additional notches, contributing to a lower fatigue strength.

The inducer drawing has been modified to specify more stringent machining and finishing requirements. New inducers were ordered.

The cause of inducer failure was attributed to a 4-per-revolution excitation. This excitation came about because of an abnormal configuration used during testing with the test rig being assembled without the impeller shroud and with the gas-bearing support installed in front of the impellers. This support has four struts.

Figure 5 shows the test configuration at the time of inducer blade failure. The curve in figure 5 shows streamlines calculated on the basis of a constant rC_u distribution along the streamlines. One expects that a core region would exit with relatively little loss in tangential momentum followed by an outer region with large turbulent mixing. The effect of the turbulent mixing could be to increase the size of the recirculating region. As can be seen, any increase in size would cause even more flow than the currently expected 15 percent to pass through the strut region. Because the flow entering the struts has a very high absolute flow angle, the struts present an almost side-on view to the flow. This should create a powerful wake causing large change of incidence at the blade leading edge and a 4-per-revolution component. The evidence suggesting that this 4-per-revolution excitation caused the failure follows.

(a) Compressor speed at failure was a 4-per-revolution first-mode resonance.

(b) The rig had been run previously without failure with both the shroud and struts removed.

(c) The presence of the struts should create a large 4-per-revolution disturbance.

A finite-element program was employed to analyze the natural frequencies and mode shapes of the inducer blade. The interference diagram is presented in figure 6. Also, the salt pattern tests have been performed on the inducer blades. The result shows that the natural frequency of the first bending mode has a band between 2800 and 3500 Hz due to blade shape variations. This frequency range has been plotted in figure 6. Including the effects of centrifugal stiffening, the inducer blade frequency coincides with a 4-per-revolution excitation at approximately 57,000 rpm. The 4-per-revolution excitation is the result of the four struts. Problems of this nature are not expected in the NASA rig since inlet struts are not employed in the inlet air path.

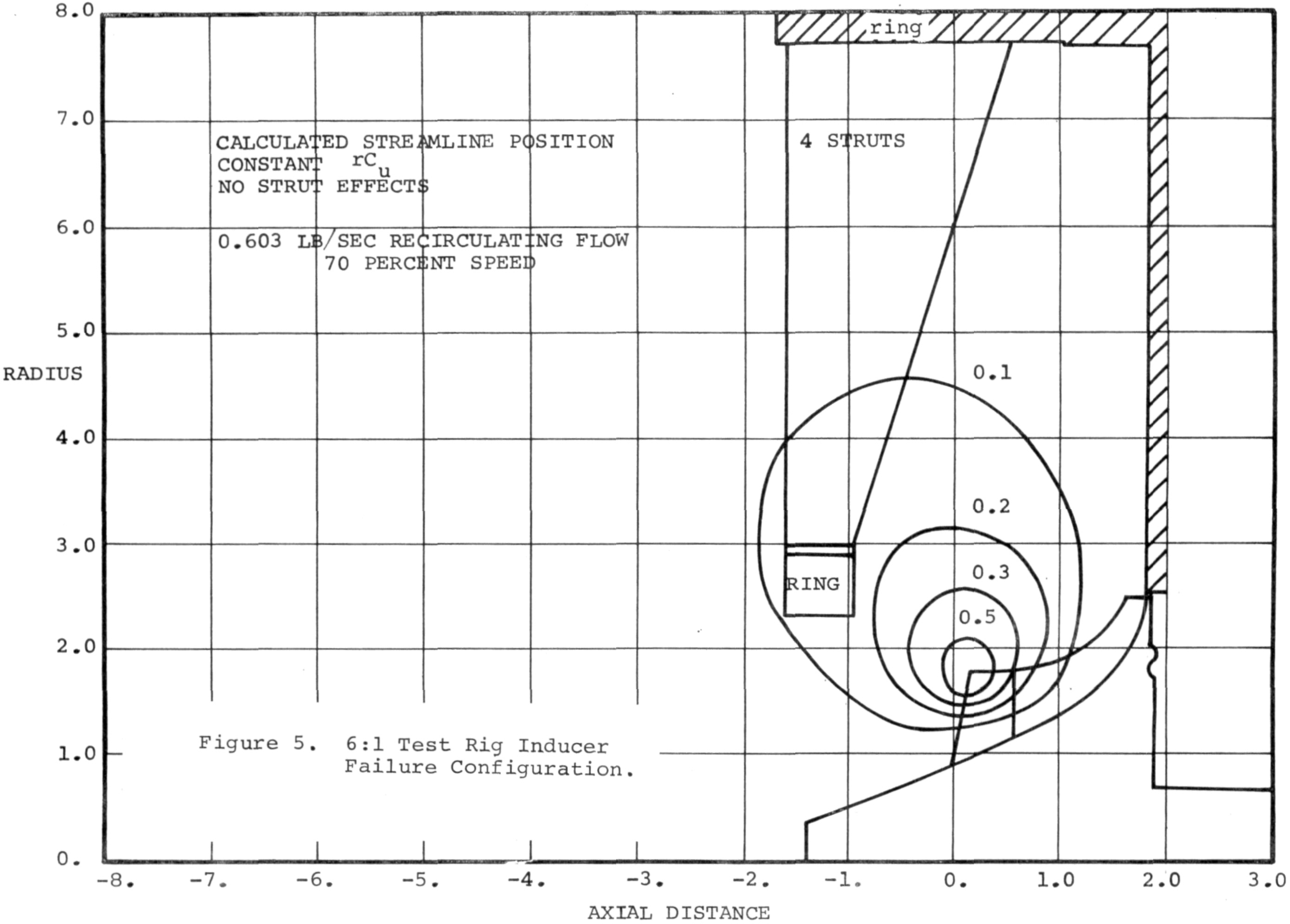


Figure 5. 6:1 Test Rig Inducer Failure Configuration.

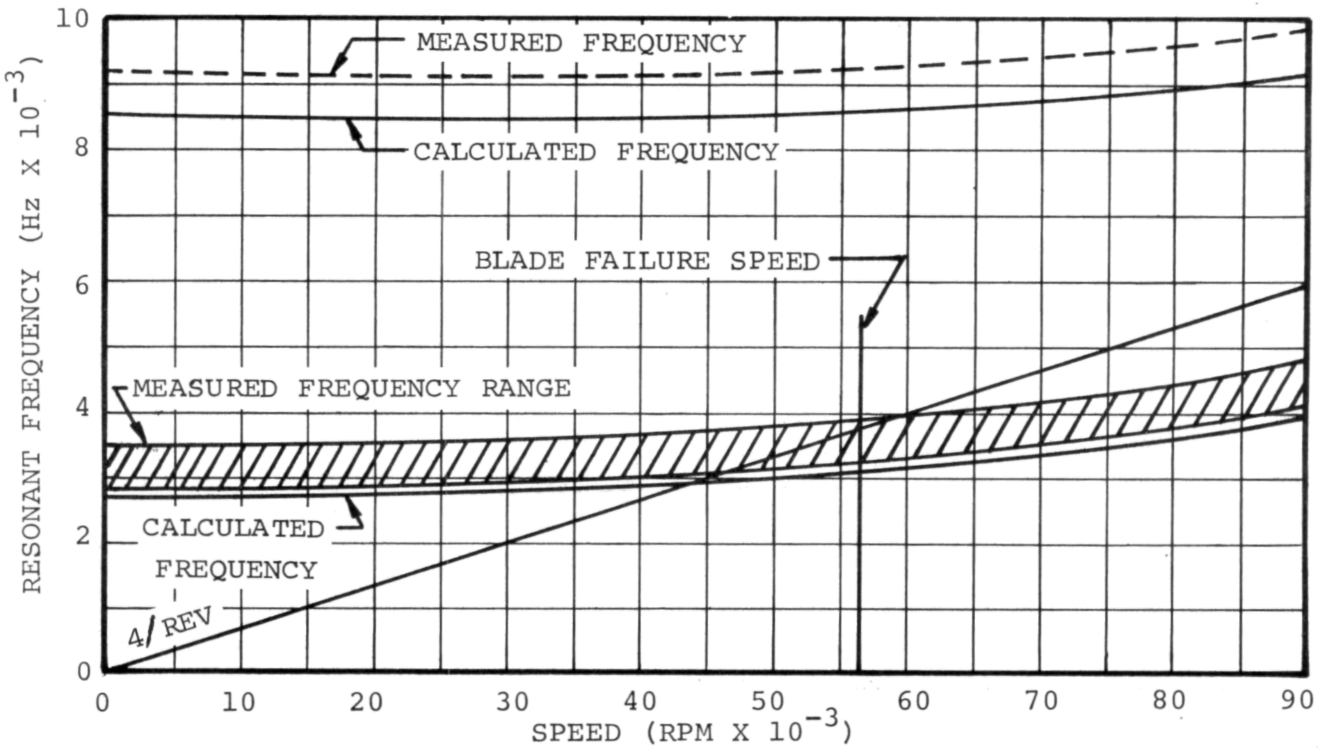


Figure 6. Compressor/Inducer Blade Interference Diagram.

APPENDIX II

ROTOR DYNAMIC ANALYSIS

(9 Pages)

APPENDIX II

ROTOR DYNAMIC ANALYSIS

The second area of concern uncovered during testing was at 65,000 rpm, the shaft excursion at the compressor end bearing increased to 2.5 mils. A critical speed problem was suspected as being the cause.

The mass and stiffness model employed in the computer analysis was checked by experimentally determining the rotor assembly's free free (laterally unsupported) natural frequency and comparing it to the calculated frequency. Agreement of the two frequencies assures the analytical elastic model is correct and the prediction of the critical speeds of the shaft supported laterally by the bearings is also correct.

The existing shaft system originally had a calculated free-free frequency of 1021 Hz and a third critical speed of 113,000 rpm. The free-free frequency was measured to 848 Hz indicating an incorrect elastic-mass model in the analysis.

Re-analysis of the shaft system resulted in a calculated free-free frequency of 821 Hz which agrees with the experimental frequency. The corrected model then resulted in a calculated third critical speed of 104,200 rpm.

The lower third critical and non-optimized hydraulic bearing clearances were considered to be the cause of the high rotor excursions. The present shaft configuration analysis is shown in figure 1 in which critical speeds versus bearing spring rate are plotted. The hydraulically mounted bearing spring rate is also shown as a function of speed. Figure 2 shows an undamped bearing load curve for a hydraulically mounted bearing system with a modified shaft system between bearings. The bearing spacer ID has been increased from 0.84 to 0.94 inch and the main shaft OD reduced from 0.72 to 0.62 inch between the bearings; this modification increases the calculated critical speed from 104,200 to 112,385 rpm.

Figure 3 depicts the mode shapes of the calculated critical speeds. The third critical speed at 112,385 rpm indicates a relatively large excursion at the bearings which provides the environment for optimum use of the hydraulically mounted bearings to provide the desired damping effects. The effects of bearing spring rate on the critical speeds are shown in figure 4.

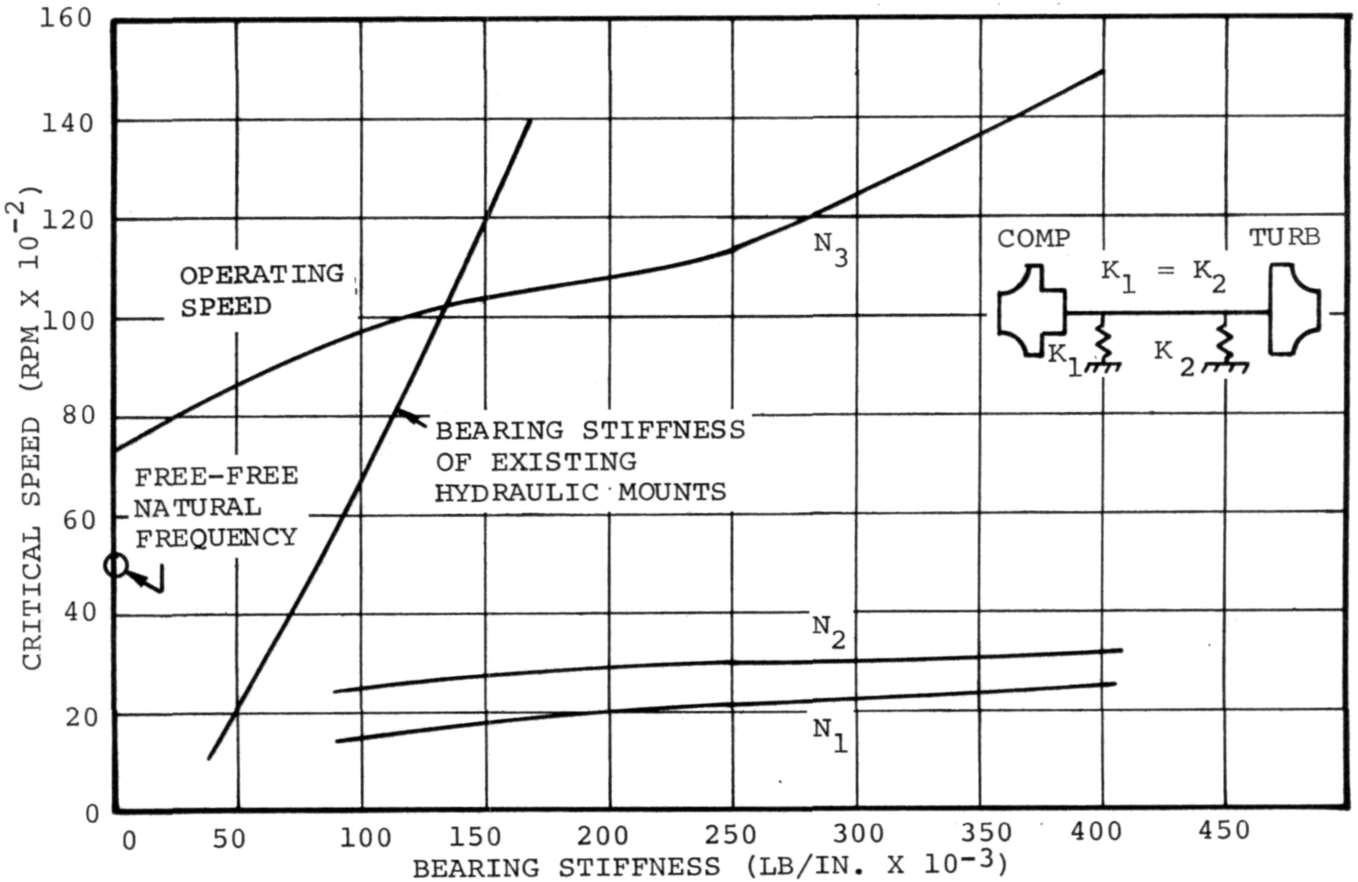


Figure 1. Critical Speeds Versus Bearing Spring Rate.

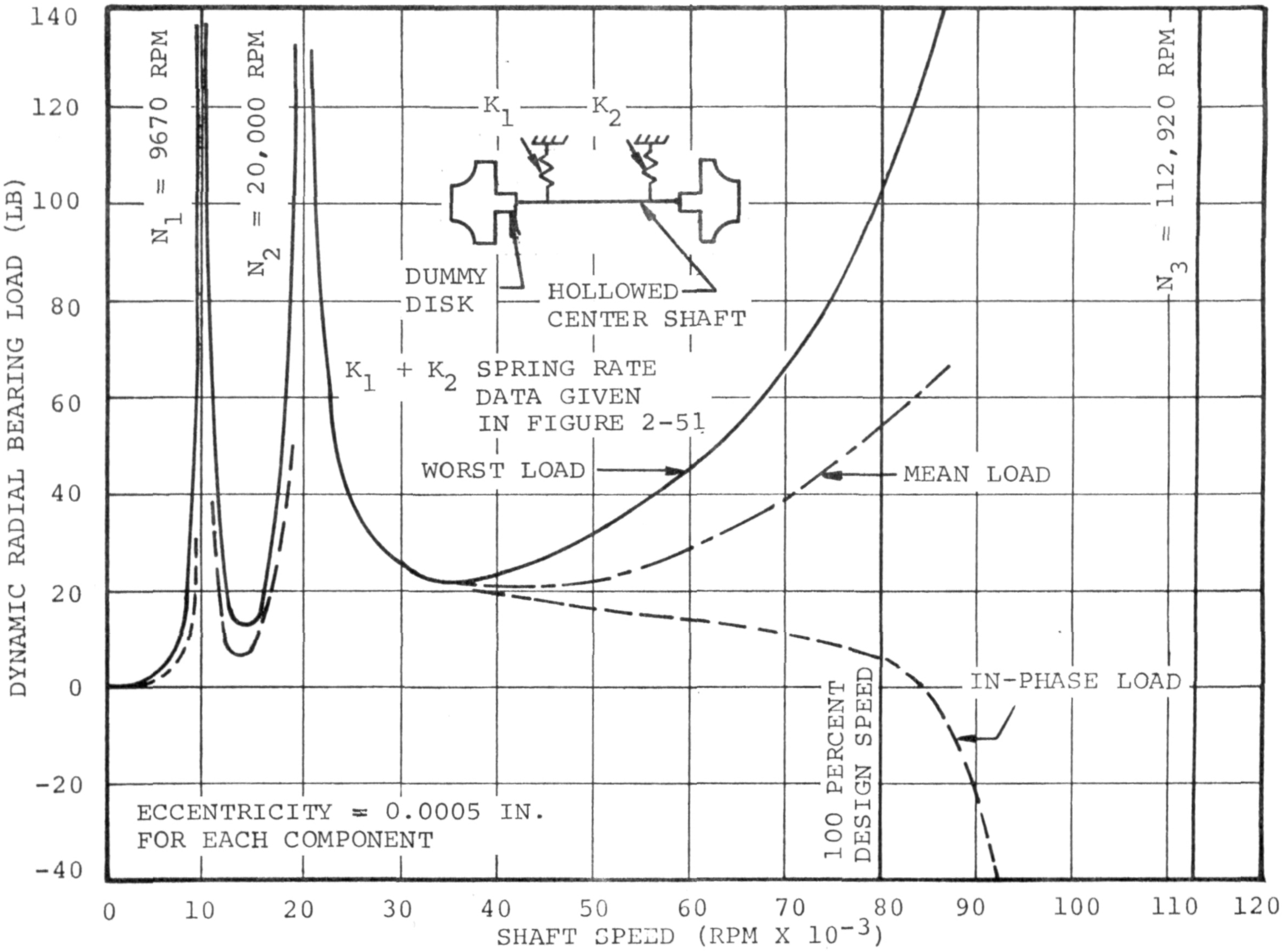


Figure 2. Dynamic Radial Bearing Load Versus Shaft Speed.

NASA 6:1 COMPRESSOR RIG

MODE SHAPE OF CRITICAL SPEEDS

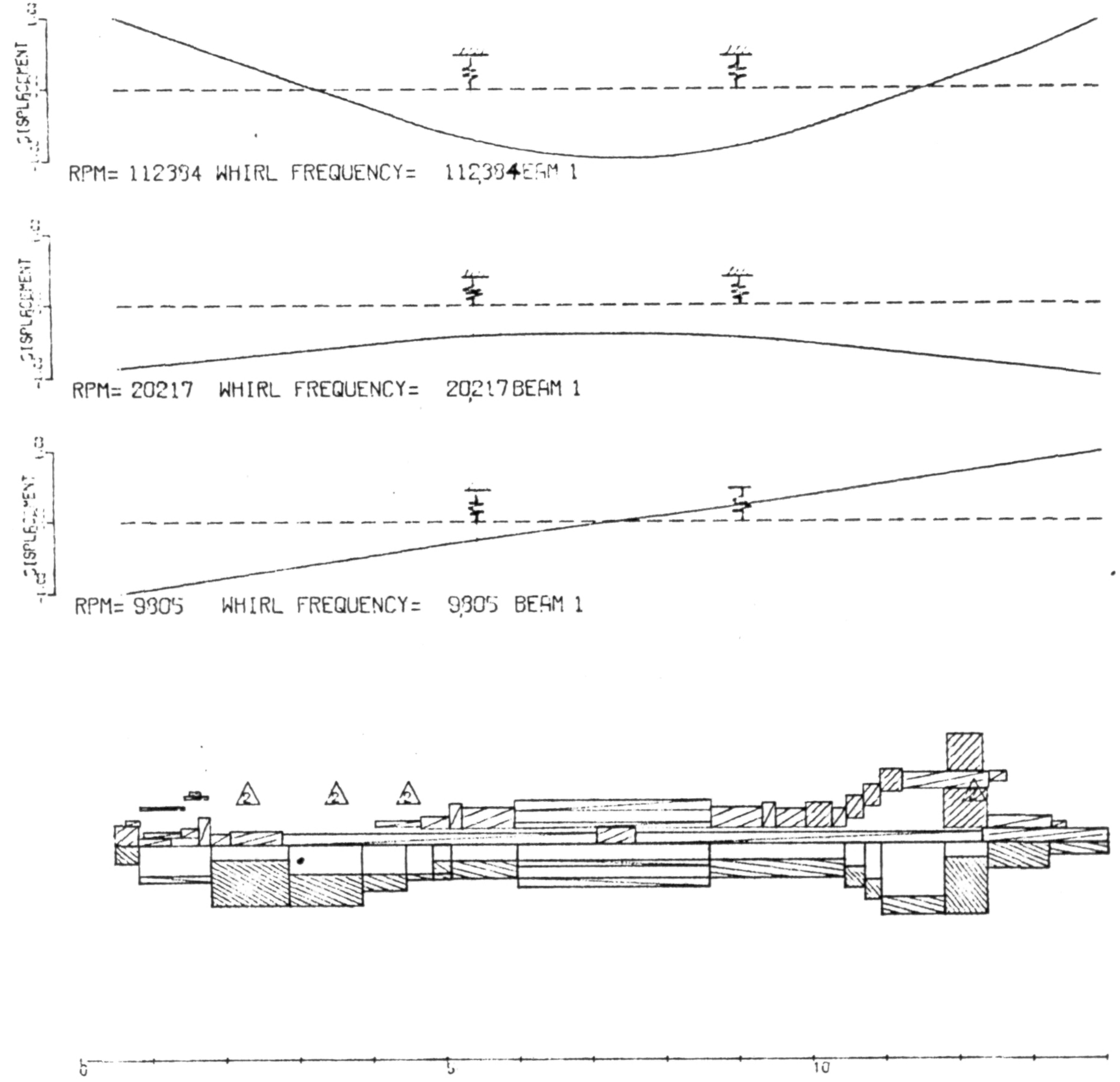


Figure 3.

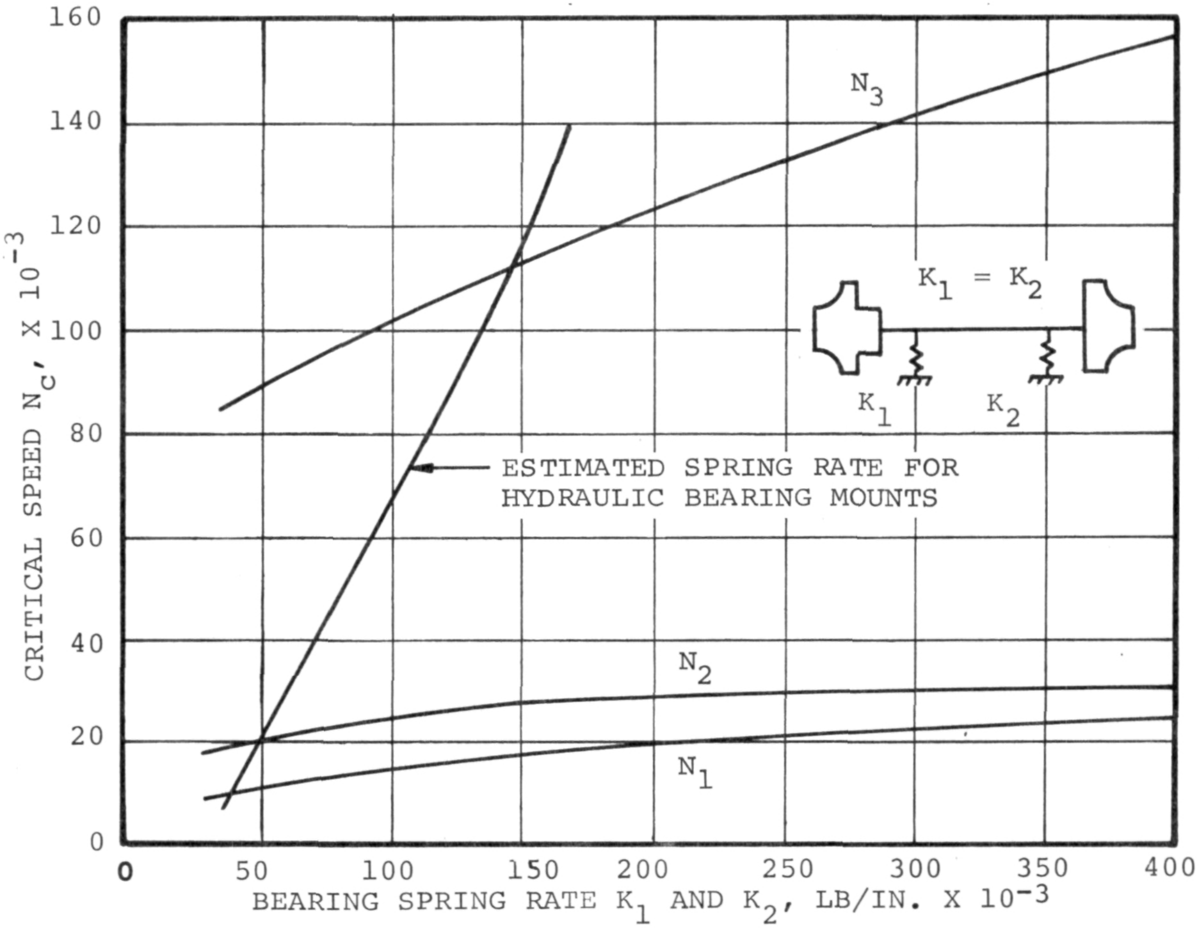


Figure 4. Critical Speeds Versus Bearing Spring Rate.

In addition to the shaft modifications, clearances were increased from 0.003 to 0.005 inch hydraulic bearing for additional damping capability.

The rig has also been analyzed for running with a dummy compressor wheel and a modified GTCP305 impeller that will be used on Contract NAS3-15328. Table I lists the inertia properties of the various wheels.

TABLE I.

Configuration		Mass in.-lb-sec^2	Ip in.-lb-sec^2	Id in.-lb-sec^2
Dummy Disks	Inducer	0.00106	0.0007	
	Impeller	0.00578	0.0112	0.00709
Designed Tandem Compressor	Inducer	0.00106	0.0007	0.0004
	Impeller	0.00567	0.0104	0.00461
Modified GTCP305 Impeller		0.00816	0.0182	0.0119

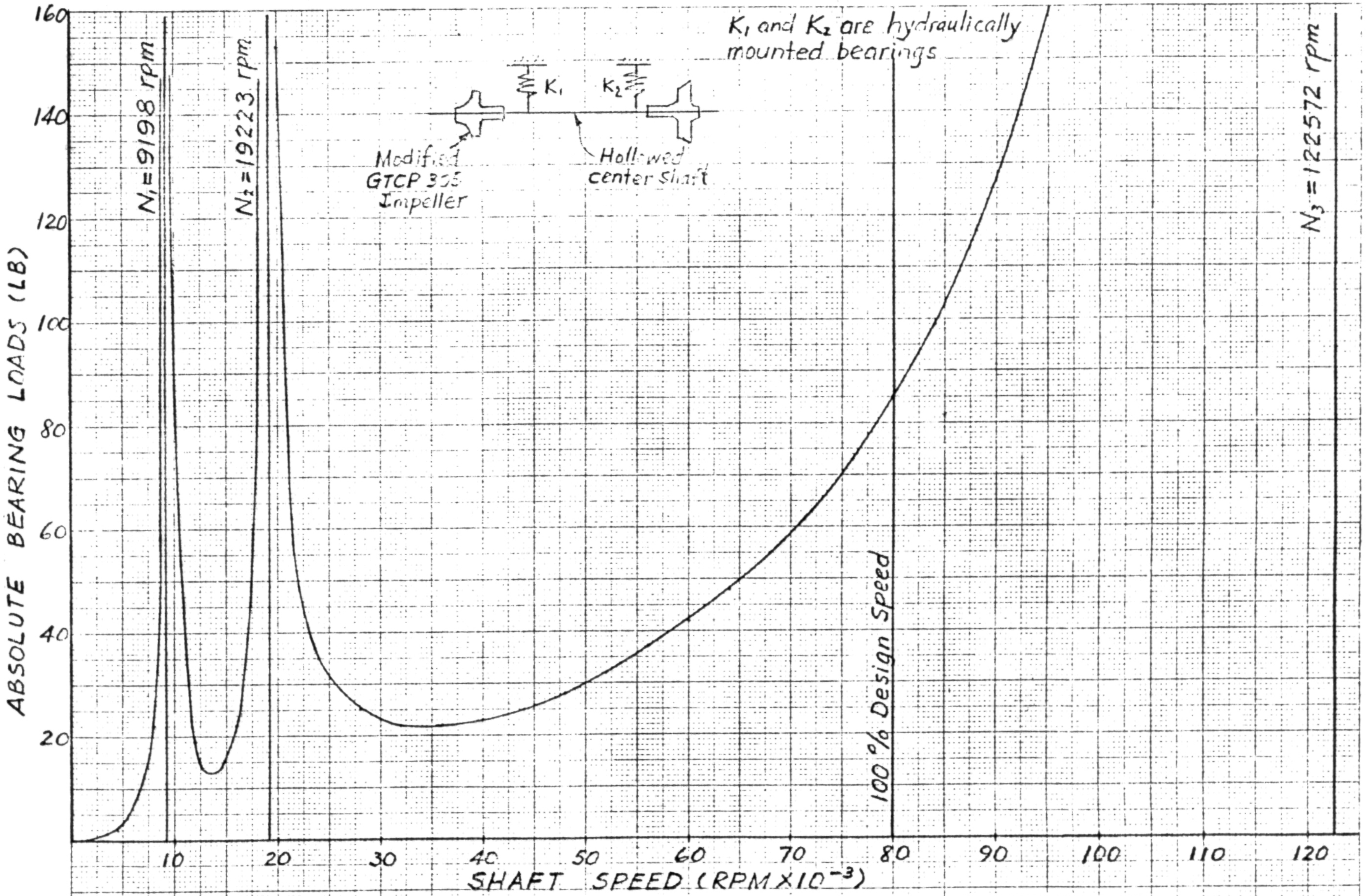
The bearing load curves for the modified 305 compressor shaft system are shown in figure 5.

The calculated third critical speed for this system is 122,572 rpm. The free-free frequency for this system is 739 Hz.

During the last rig test it was also observed that the turbine side bearing had spun in its housing. In order to preclude the problem of spinning, the bearing has been repinned as originally designed.

Retesting of the test rig with the dummy masses is scheduled for mid-March 1972.

A crack was observed in a blade on the NASA turbine wheel and is shown on figure 6. The wheel was returned to NASA-Lewis for examination.



CALCULATED BY			NASA 6:1 COMPRESSOR RIG BEARING LOAD VS. SHAFT SPEED	Figure 5
TRACED BY				
CHECKED BY				
APPROVED BY			AiResearch Manufacturing Company of Arizona	
UNIT NO.				

F5105

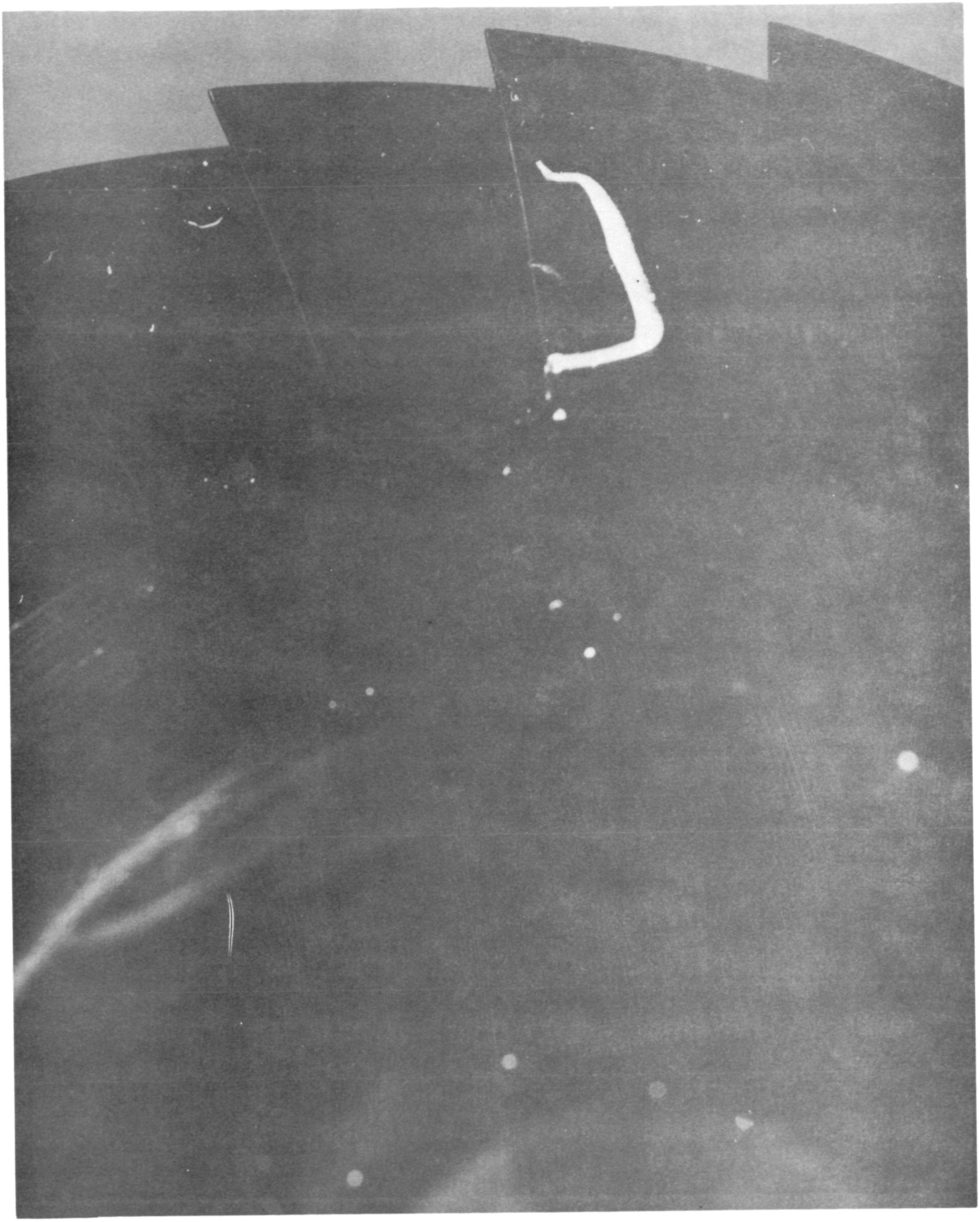


Figure 6. Blade Crack, NASA Turbine Wheel.

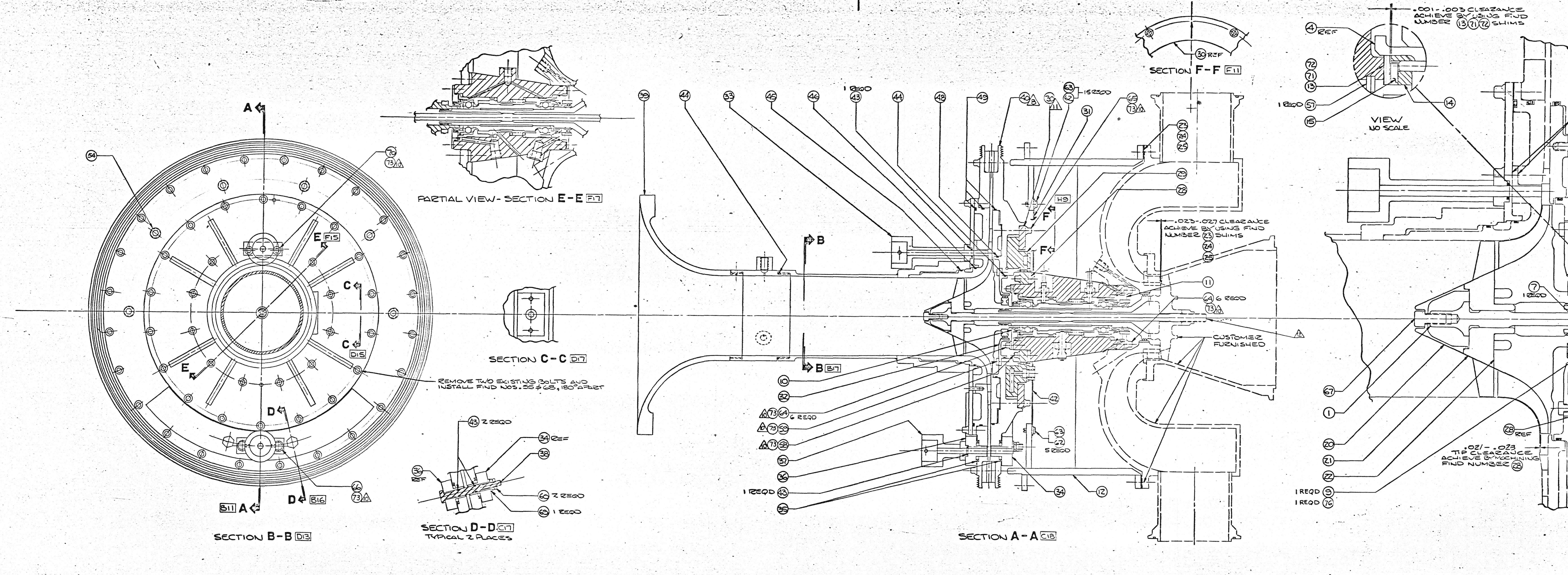
APPENDIX III

COMPRESSOR RESEARCH PACKAGE
ASSEMBLY DRAWINGS

(3 Drawings)

REVISIONS

ZONE	LTR	DESCRIPTION	DATE	APPROVED
2M	A	FIND 53 WAS NAS1101C6-6	3-24-71	CHE
	B	ADDED FIND 54 TO DWG	6-4-71	CHE
7C	C	.021-.023 WAS .007-.011	7-23-71	[illegible]
6/M	D	DELETED ITEM #47 WAS S8990-47 PACKING 1 REQD	9-24-71	CHE
	E	ADDED ITEMS 76 & 77 ADDED NOTE #12	11-1-72	W. Sharp
4/M G16	F	FIND NO. 39 WAS P/N SKP25708-1. FIND NO. 39 UPDATED PICTORIALLY.	11-30-72	W. Sharp



12. SHAFT STRETCH TENSION LOAD 11,700 - 12,300 LB.
11. INSTALL FIND No. 30 SO THAT ARROW IS VISIBLE AS SHOWN IN SECTION F-F
10. LOCKWIRE PER MS 33540
9. TORQUE VALUES FOR THREADED FASTENERS SHALL BE IN ACCORDANCE WITH AIRESEARCH AF 5349
8. DISASSEMBLE FIND NUMBER 40 AND INSTALL FIND NUMBERS 45, 46, 48 AND 49. APPLY FIND No. 56 TO SURFACE INDICATED. REPLACE EXISTING BOLTS WITH FIND No. 65
7. MANUFACTURED BY ELECTRO PRODUCTS CORPORATION
6. MANUFACTURED BY BENTLY NEVADA CORPORATION MINDEN, NEVADA
5. VENDOR ITEM: SEE APPLICABLE SPECIFICATION OR SOURCE CONTROL DRAWING
4. PACKAGE AS MATCHED SET WITH PACKAGE LABELED "KEEP IN MATCHED SET. DO NOT USE SEPARATELY."
3. IDENTIFY AND SERIALIZE AS A MATCHED SET PER AIRESEARCH MC5014 CLASS II A.
2. MANUFACTURE, INSPECT, STOCK AND SELL AS REPLACEMENT PARTS IN MATCHED SETS ONLY
1. SHALL CONFORM TO AIRESEARCH: SC 6000 INTERPRETATION OF DRAWING SC 6001 MACHINED FEATURES

PARTS LIST

QUANTITY REQD	FIND NO	PART NO OR IDENTIFYING NO	SYM	NOMENCLATURE OR DESCRIPTION	CODE IDENT	MATERIAL AND SPECIFICATION	UNIT WT	ZONE
AR	77	S8990-125		PACKING				C8 C4
2	76	SKP32204		SPACER				C8 C4
3	75	#3016	7	SPEED PICKUP				D5
4	74	#306L-36-.80	6	PROBE				F6
AR	73	MS20995C20		LOCKWIRE				G10
AR	72	SKP25649-3		SHIM				H9
AR	71	SKP25649-2		SHIM				G9
2	70	MS21288-16		BOLT, MACHINE				G17
2	69	AN960C416		WASHER				C15
2	68	AN315-3C		NUT, LOCK				D15
1	67	MS21295-30		SCREW, SELF-LOCKING				D8
2	66	MS21289-08		BOLT, MACHINE				C18
18	65	MS21288-20		BOLT, MACHINE				G10
12	64	MS21288-06		BOLT, MACHINE				[illegible]
20	63	AN3C-3		BOLT, MACHINE				G11
20	62	MS35338-138		WASHER, LOCK				G11
3	61	MS9361-12		NUT				C5
8	60	MS9361-10		NUT				F5 F6
8	59	NAS1101C3-6		SCREW, MACHINE				D13
8	58	MS21288-12		BOLT, MACHINE				C13
2	57	626-001-9113	5	SNAP RING				F5 G6
AR	56	219-189-9001	5	COMPOUND				G6
2	55	137-501-9001	5	BEARING, ROD END				D15
6	54	AN932-S3		PLUG, PIPE				G19
1	52	S9413-40		PACKING				F5
1	51	S9413-34		PACKING				C6
1	50	S9413-33		PACKING				F5
3	49	S8990-274		PACKING				G11
1	48	S8990-161		PACKING				G12
	47							
1	46	S8990-47		PACKING				G13
1	45	S8990-46		PACKING				G13
2	44	S8990-44		PACKING				G12
6	43	S8990-9		PACKING				G13
1	42	SKP25665-2		BAFFLE				D11
2	41	976693-1		BEARING				F5
1	40	SKP25710-1		MATCHED SET, DIFFUSER				G11
1	39	SKP26477-1		BELL MOUTH				G14
2	38	SKP25679-1		BOLT, SPECIAL				C15
1	37	SKP25678-1		MOUNT, PROBE				C13
1	36	SKP25677-1		MOUNT, SLIDE				C13
2	35	SKP25676-1		SEAL				C13
1	34	SKP25675-1		MOUNT, SLIDE				C11
1	33	SKP25670-1		MOUNT, PROBE				G14
1	32	SKP25669-1		PLATE, SEAL				D13
3	31	SKP25665-1		BAFFLE				G11
1	30	SKP25662-1		RING, AIR BEARING				G11
1	29	SKP25661-1		BEARING ASSY, AIR				F10
1	28	SKP25660-1		SPACER				F10
AR	27	SKP25652-3		SHIM				F4
AR	26	SKP25652-2		SHIM				F4
AR	25	SKP25659-3		SHIM				G10
AR	24	SKP25659-2		SHIM				G10
AR	23	SKP25659-1		SHIM				G10
1	22	SKP25658-1		WHEEL				C8
1	21	SKP25657-1		INDUCER				C8
1	20	SKP25656-1		CONE, INLET				C8
1	18	SKP25654-1		NUT, TIE BOLT				D3
1	17	SKP25653-1		SPRING				C5
AR	16	SKP25652-1		SHIM				F4
1	15	SKP25651-1		SPRING, WASHER				G8
1	14	SKP25650-1		RETAINER ASSEMBLY				G7
AR	13	SKP25649-1		SHIM				G9
1	12	SKP25648-1		HOUSING ASSEMBLY				C10
1	11	SKP25647-1		RETAINER ASSY, SEAL				E9
1	10	SKP25646-1		RETAINER ASSY, SEAL				D13
2	9	PAP203948		SPRING, SEAL				[illegible]
2	8	PAP203946		SEAL				[illegible]
2	7	PAP203945		SEAL				[illegible]
2	6	SKP25645-1		ROTOR, SEAL				[illegible]
1	5	SKP25644-1		RETAINER				C5
1	4	SKP25643-1		RETAINER				F5
1	3	SKP25642-1		SPACER				F5
1	2	SKP25641-1		SHAFT				C4
1	1	SKP25640-1		TIE BOLT				D8

-1 PART NUMBER — PARTS LIST

REDUCED SIZE

REQD	NEXT ASSY	USED ON
(-1)	SKP25713 TEST	

HEAT TREATMENT — PROCESS

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

COMPRESSOR & TURBINE ASSEMBLY, MATCHED SET, TEST RIG

SIZE J — CODE IDENT NO 99193 — DWG NO SKP25711

DESIGN ACTIVITY APVD 7-16-71

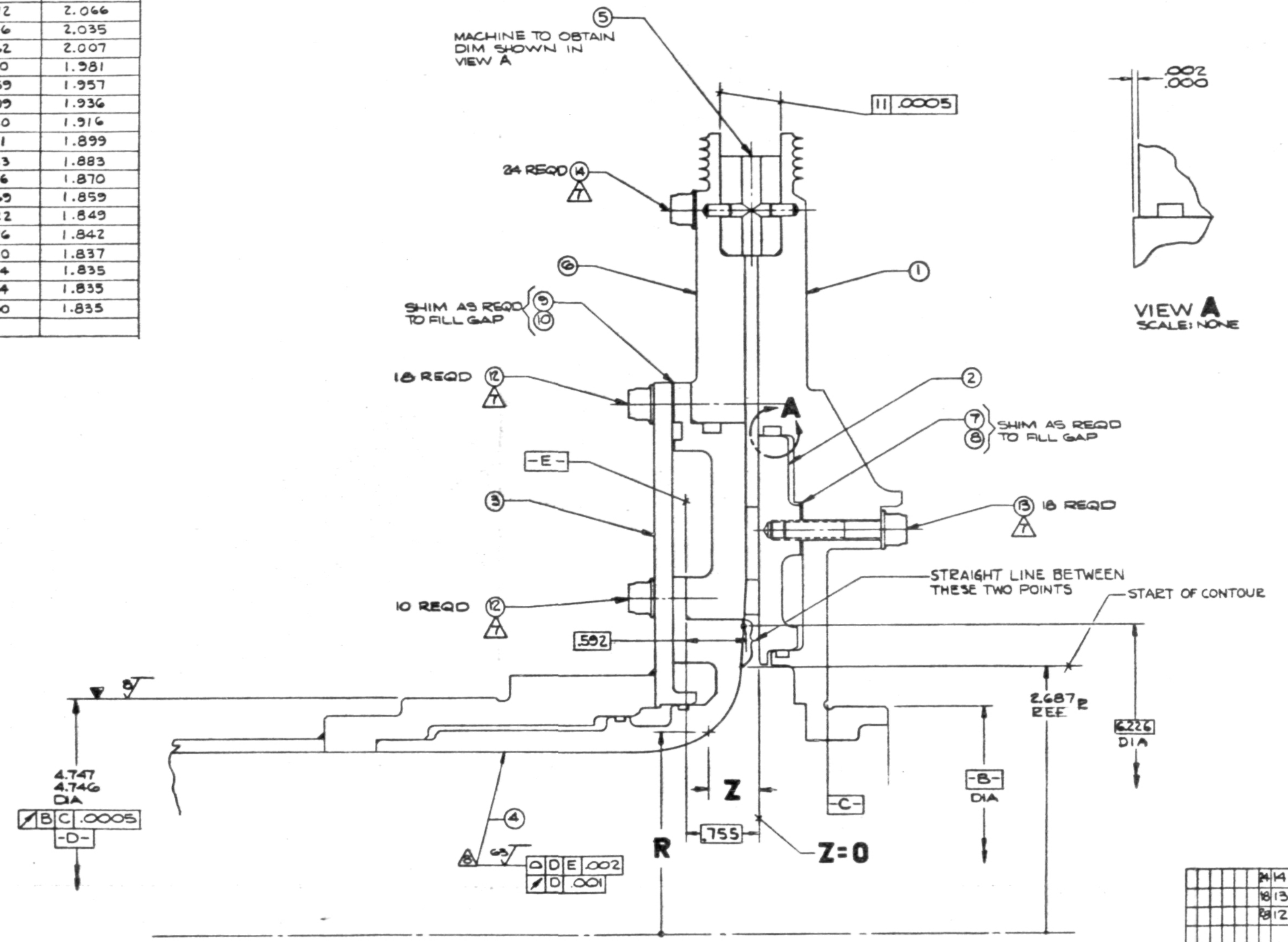
MICROFILMED

SKP25711 E

ZONE	LTR	DESCRIPTION	DATE	APPROVED
F4 C3 D5	A	DELETED .621-.631 DIM 6.226 BASIC DIA WAS 6.215 BSC DIA .592 BSC WAS .591 BSC	3-22-71	SHE

BASIC CONTOUR COORDINATES

FROM Z	R RAD
.189	2.687
.189	2.650
.189	2.623
.191	2.568
.196	2.514
.205	2.460
.218	2.408
.236	2.357
.259	2.307
.285	2.261
.316	2.216
.351	2.174
.389	2.136
.429	2.099
.472	2.066
.516	2.035
.562	2.007
.610	1.981
.659	1.957
.709	1.936
.760	1.916
.811	1.899
.863	1.883
.916	1.870
.969	1.859
1.022	1.849
1.076	1.842
1.130	1.837
1.184	1.835
1.234	1.835
1.800	1.835



7. TORQUE VALUES FOR THREADED FASTENERS SHALL BE IN ACCORDANCE WITH AIRESEARCH AF5349

6. PACKAGE AS MATCHED SET WITH PACKAGE LABELED, "KEEP IN MATCHED SET. DO NOT USE SEPARATELY."

5. IDENTIFY & SERIALIZE AS MATCHED SET PER AIRESEARCH MC5014, CLASS II A.

4. FINISH ALL OVER 125√ EXCEPT AS NOTED.

3. ON SURFACE INDICATED THUS ▼: HARD CHROME PLATE .002-.005 THICK PER SPEC QQ-C-320, CLASS 2.

2. MANUFACTURE, INSPECT, STOCK & SELL AS REPLACEMENT PARTS IN MATCHED SETS ONLY.

1. SHALL CONFORM TO AIRESEARCH: SC6000 INTERPRETATION OF DRAWINGS SC6001 MACHINED FEATURES

10. SEE L687509 FOR TOOLING LAYOUT

9. MACHINING SEQUENCE:
 a. ESTABLISH DIM IN VIEW A & SHIMMING
 b. ESTABLISH 4.7465 DIA REF
 c. ESTABLISH SHROUD CONTOUR ON (4)

8. AERODYNAMIC SHAPE TO BLEND SMOOTHLY IN ALL DIRECTIONS WITHIN THE CONTOUR TOLERANCE. NO FLATS OR SHARP PEAKS ALLOWED

DATUM	B	C	D	E
LOCATION	C4	C4	C7	E5

LAST LTR E
ALSO USED ZA R

PARTS LIST

QUANTITY REQD	FIND NO.	PART NO. OR IDENTIFYING NO.	SYM	NOMENCLATURE OR DESCRIPTION	CODE IDENT	MATERIAL AND SPECIFICATION	ZONE
24	14	MS21288-17		BOLT, MACHINE			F5
18	13	MS21288-15		BOLT, MACHINE			D3
28	12	MS21288-08		BOLT, MACHINE			E5 D5
A/R	10	SKP25819-2		SHIM			E5
A/R	9	SKP25819-1		SHIM			E5
A/R	8	SKP25781-2		SHIM			E4
A/R	7	SKP25781-1		SHIM			E4
1	6	SKP25709-1		HOUSING, SEAL			F5
1	5	SKP25674-1		PLATE			G5
1	4	SKP25668-1		SHROUD, COMP. INLET			C5
1	3	SKP25666-1		INLET HSG. ASSY			D5
1	2	SKP25664-1		DIFFUSER ASSY			E4
1	1	SKP25663-1		HOUSING, AIR BRG			F4

-1 ← PART NUMBER

REQD	NEXT ASSY	USED ON
(K-1)	SKP25714	TEST

HEAT TREATMENT: SEE NOTE 3

SIGNATURES / DATES: 2-12-71, 2-18-71, 3-18-71, 3-15-71, 3-18-71, 3-18-71

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

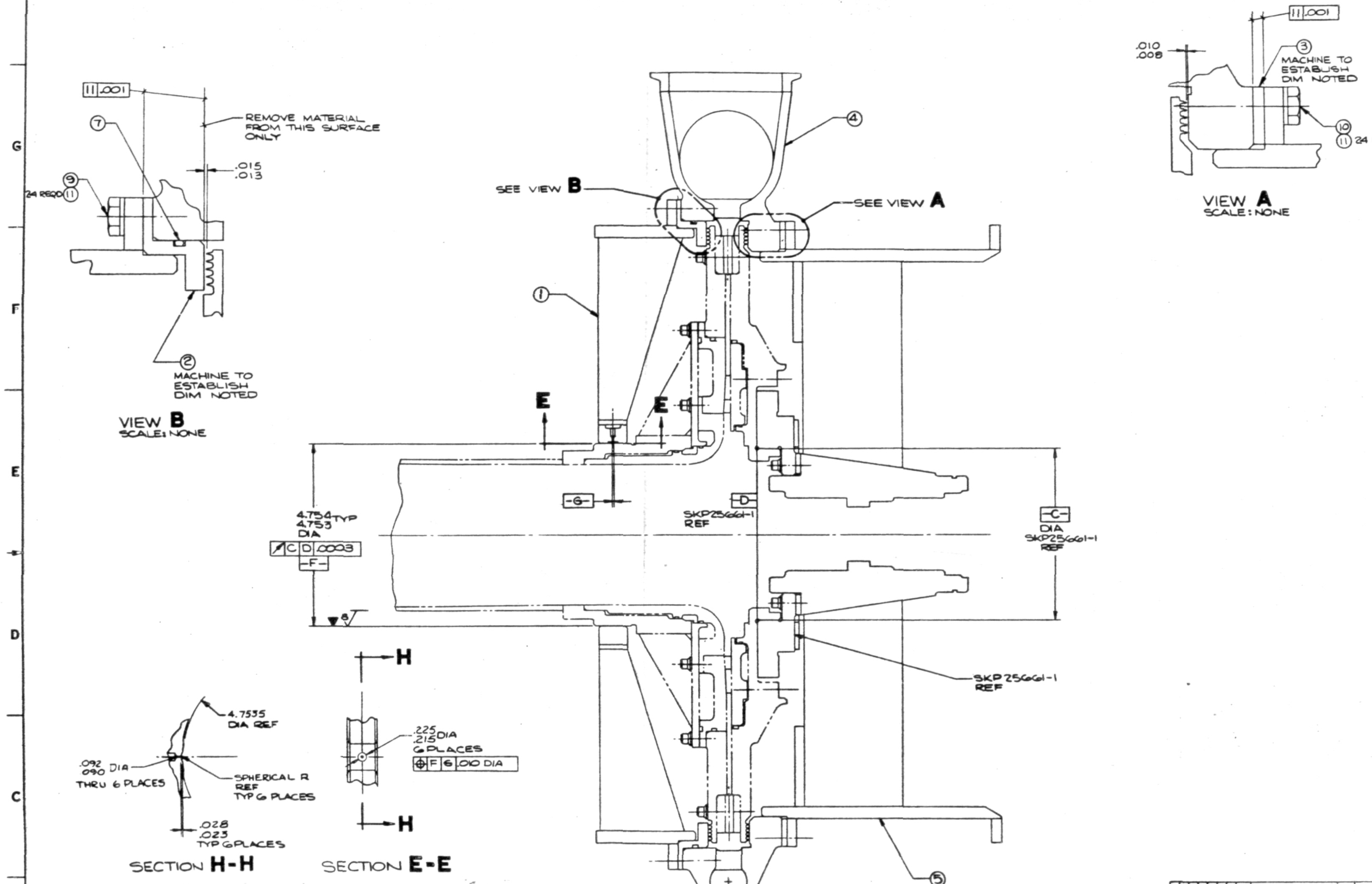
DWG TITLE: HOUSING MATCHED SET, DIFFUSER

SIZE E | CODE IDENT NO. 99193 | DWG NO. SKP25710

SCALE 2/1 & NOTED | WT | SHEET — OF —

SKP25710

REVISIONS				
ZONE	LTR	DESCRIPTION	DATE	APPROVED
C8 C7	A	.015-.017 TYP 6 PLACES -WAS- .020-.010 TYP 6 PLACES SPHERICAL AIR FLOW DEPRESSIONS WERE RECTANGULAR		
E7 C8	B	4.754-4.753 WAS 4.751-4.750 .028-.023 WAS .017-.015 ADDED .092-.090 DIA THRU 6 PLACES	3/23/72	
	C	IN VIEW "A" .008-.010 WAS .003-.005 IN VIEW "B" .013-.015 WAS .003-.005	11-1-72	W. Sharp



7. TORQUE VALUES FOR THREADED FASTENERS SHALL BE IN ACCORDANCE WITH AIRESEARCH AF 5349

6. PACKAGE AS MATCHED SET WITH PACKAGE LABELED, "KEEP IN MATCHED SET. DO NOT USE SEPARATELY."

5. IDENTIFY & SERIALIZE AS MATCHED SET PER AIRESEARCH MC5014, CLASS IIA

4. FINISH ALL OVER 125√ EXCEPT AS NOTED

3. ON SURFACE INDICATED THUS ▼: HARD CHROME PLATE .002-.005 THICK PER SPEC QQ-C-320, CLASS 2.

2. MANUFACTURE, INSPECT, STOCK & SELL AS REPLACEMENT PARTS IN MATCHED SETS ONLY.

1. SHALL CONFORM TO AIRESEARCH: SCG000 INTERPRETATION OF DRAWING SCG001 MACHINED FEATURES

8. ASSEMBLY-MACHINING SEQUENCE:
   a. ESTABLISH DIMENSION IN VIEW A
   b. ESTABLISH DIMENSION IN VIEW B
   c. ESTABLISH 4.7505 DIA REF WITH ITEMS IN PHANTOM REMOVED.

QUANTITY REQD	FIND NO	PART NO. OR IDENTIFYING NO.	SYM	NOMENCLATURE OR DESCRIPTION	CODE IDENT	MATERIAL AND SPECIFICATION	ZONE
48	11	AN960C416		WASHER, FLAT			G8 G2
24	10	AN4C7		BOLT, MACHINE			G2
24	9	AN4C6		BOLT, MACHINE			G8
1	7	S8990-282		PACKING			G8
1	5	SKP25711-1		COMP. & TURB ASSY			C4
1	4	SKP25673-1		COLLECTOR, COMP			G4
1	3	SKP25672-1		SPACER			H2
1	2	SKP25671-1		SPACER, SEAL			F8
1	1	SKP25667-1		HOUSING, AIR BRG			F6
-1	← PART NUMBER			PARTS LIST			

DATUM	C	D	F	G	LAST LTR	H
LOCATION	E3	E5	D7	E6	ALSO USED	

1(-1)	FINAL	TEST RIG
REQD	NEXT ASSY	USED ON
HEAT TREATMENT	PROCESS	
	SEE NOTE 3	

SIGNATURES / DATES: 2-17-71; 2-23-71; 2-25-71; 2-25-71

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX ARIZONA

DWG TITLE
TEST RIG ASSEMBLY, COMPRESSOR, MATCHED SET

SIZE E | CODE IDENT NO. 99193 | DWG NO. SKP25775

SCALE 1/1 & NOTED | WT | SHEET — OF —

MICROFILMED